UNIVERSITI TEKNOLOGI MALAYSIA

BORANG PENGESAHAN
LAPORAN AKHIR PENYELIDIKAN

TAJUK PROJEK : DESIGN AND DEVELOPMENT OF ROTATING SLEEVE

REFRIGERANT COMPRESSOR

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DESIGN AND DEVELOPMENT OF ROTATING-SLEEVE REFRIGERANT COMPRESSOR

(REKABENTUK DAN PEMBANGUNAN PEMAMPAT REFRIGERAN SILINDER BERPUTAR)

PROF DR MD NOR MUSA

RESEARCH VOTE NO:
71811

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ABSTRACT

A Single Vane Rotating Sleeve compression concept a patent of which has been filed by UTM, is the main focus of this study. Research and development on the concept was carried out to investigate its feasibility on a refrigeration system. The concept was designed based on the specifications of the existing reciprocating compressor installed in a refrigerator. In order to design a functional prototype, the author has conducted literature study on existing rotary compressor models such as rolling piston and sliding vane types that are used in room and car air conditioning systems respectively. The literature study is crucial in areas such as the geometrical optimization, material selection, tolerance and surface finishing in designing the prototype. Preliminary concept development and design are also conducted for the new concept investigating critical design data for used in future research. Further research and development however is needed to improve the compressor performance up to a commercial acceptable level.

ABSTRAK

Konsep pemampatan *Single Vane Rotating Sleeve* yang telah dipatenkan oleh UTM merupakan fokus utama kajian ini. Kajian dan pembangunan telah dilaksanakan ke atas konsep berkenaan untuk mengenalpasti kebolehfungsiannya di dalam sistem pendinginan. Konsep yang telah direkabentuk adalah berdasarkan spesifikasi asal pemampat salingan yang dipasang oleh kilang di dalam peti sejuk. Untuk menghasilkan satu prototaip yang baik, penulis telah mengkaji beberapa model pemampat berputar yang sedia ada seperti jenis *rolling piston* dan *sliding vane* yang masing-masingnya digunakan dalam sistem penyamanan udara bilik dan kereta. Kajian ilmiah amat penting terutama dalam aspek geometri, pemilihan bahan, kelegaan dan kelicinan permukaan dalam merekabentuk prototaip. Langkah awal pembangunan konsep turut dijalankan untuk konsep baru tersebut bagi mendapatkan data penting yang akan digunakan pada penyelidikan akan datang. Walaubagaimanapun, kajian dan pembangunan yang lebih mendalam perlu dilakukan untuk memajukan prestasi pemampat ke arah kesesuaian untuk dikomersialkan.

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LIST OF SYMBOLS

A	-	Area
COP	-	Coefficient of performance
D	-	Inner sleeve diameter
d	-	Rotor diameter
e	-	Eccentric distance
h	-	Enthalpy
l	-	Length
m	-	Mass
$\dot{m}$	-	Mass flowrate
N	-	Speed
n	-	Polytropic compression
p	-	Compressor power
p_1	-	Suction Pressure
p_2	-	Discharge Pressure
$\dot{Q}$	-	Refrigeration effect
$\dot{Q}$	-	Refrigeration capacity
R	-	Gas constant
R	-	Inner sleeve radius
r	-	Rotor radius
s	-	Entropy
T	-	Temperature
T_1	-	Suction Temperature
T_2	-	Discharge Temperature
t	-	Height
V	-	Volume
V_1	-	Total volume
V_2	-	Discharge volume

V_3	-	Clearance volume
V_s	-	Swept volume
v	-	Velocity
W	-	Compressor work
W_{12}	-	Compression work
w	-	Width of vane
β	-	Sleeve rotation angle
θ	-	Rotor rotation angle
ρ	-	Density
π	-	Pi
η_{com}	-	Compressor
η_{cp}	-	Compression
η_{mec}	-	Mechanical
η_{mot}	-	Motor efficiency
η_v	-	Volumetric efficiency
Δ	-	Area
Φ	-	Diameter

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CHAPTER 1

INTRODUCTION

1.1 Fundamental of Refrigeration

Refrigeration is the process of removing heat from a space or substance and transfer that heat to another space or substance. The term refrigeration is used here to include both the cooling process to preserve food and comfort cooling (air conditioning). In any refrigerating process, the substance employed as the heat absorber or cooling agent is called the refrigerant. The refrigerant absorbs heat by evaporating at low temperature and pressure and remove heat by condensing at a higher temperature and pressure. As the heat is removed from the space, the area appears to become cooler. The process of refrigeration occurs in a system which comprises of a compressor, a condenser, a capillary and an evaporator arranged as depicted schematically in Figure 1.1.

Compressor is a mechanical device to compress and pump the refrigerant vapour from a low-pressure region (the evaporator) to a high-pressure region (the condenser). The condenser is a device for removing heat from the refrigeration system. In the condenser, the high temperature and high-pressure refrigerant vapour transfers heat through the condenser tube wall to the surrounding. When the temperature of the refrigerant vapour reaches the saturation level, the latent heat is released causes condensation process and the refrigerant vapour changes phase to a liquid form. The metering device (throttle valve or capillary tube) controls the refrigerant flow from the condenser to the evaporator and separates the system to high pressure and low-pressure sides. The evaporator is a device for absorbing heat

from the refrigerated space into the refrigeration system by evaporating the refrigerant [1, 2].

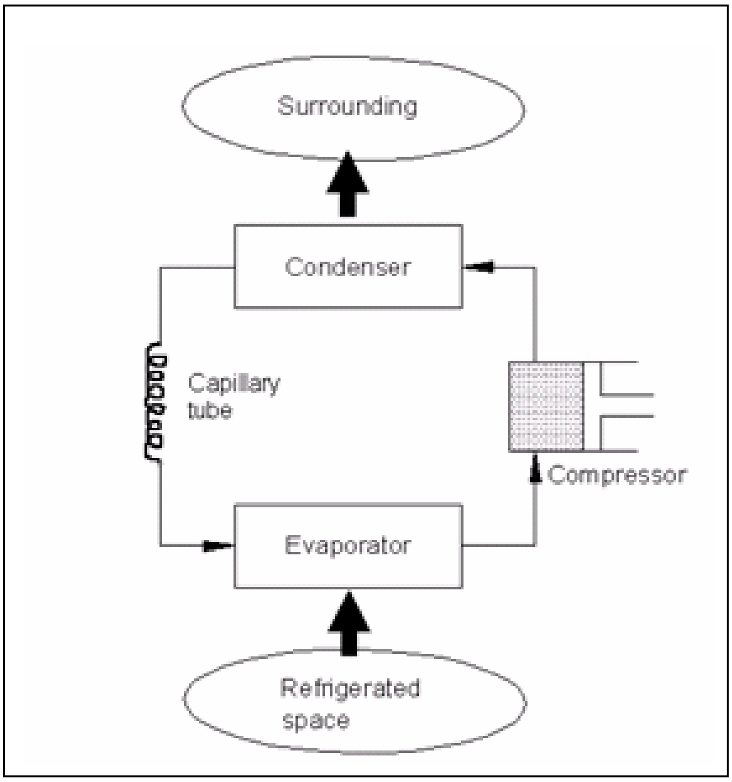


Figure 1.1: Schematic diagram of refrigeration system

1.2 Refrigerating Compressor

Refrigerating compressor is a heart of a refrigeration system. It raises the pressure of the refrigerant so that it can be condensed into liquid, throttled to a lower pressure, and evaporated into vapour to produce the refrigeration effect. It also provides the primary force to circulate the refrigerant through the cycle [3].

According to the compression process, the refrigerating compressor can be divided into two main classifications and each classification can be further sub-divided into several groups, as illustrated in Figure 1.2. The positive displacement compressor is a type that increases the gas pressure by reducing the internal volume of the compression chamber through the mechanical force that is applied to the compressor. Whereas, a non-positive displacement compressor is where the compression of the gas depends mainly on the conversion of dynamic pressure into static pressure [4].

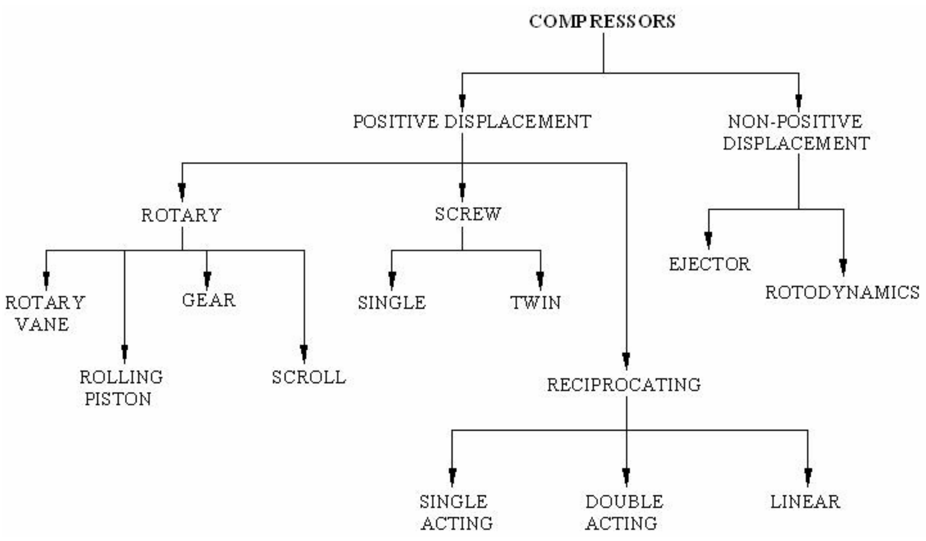


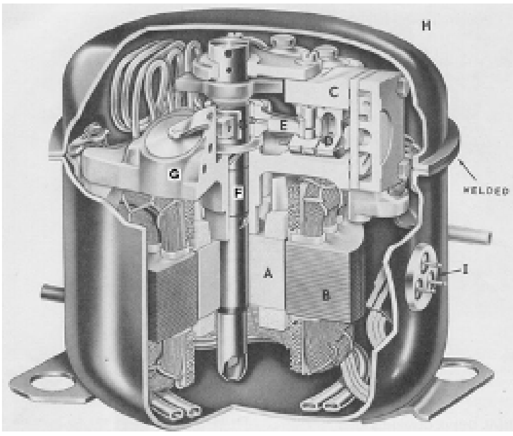
Figure 1.2: Classification of compressor

The positive displacement compressors can be further classified into rotary, screw and reciprocating types. Only rotary and reciprocating types are discussed in this report. These two types of compressors are packaged into three different assemblies as described in the following discussion:

i. **Hermetic compressor**. In the assembly the motor and the compressor are sealed or welded in the same housing as shown in Figure 1.3. Hermetic compressor has two advantages that it minimizes leakage of refrigerant and has mechanism to cool the motor by using the suction vapour flowing through the motor windings. Motor windings in hermetic compressors must be compatible with the refrigerant and lubricating oil, resist the abrasive effect of the suction vapour, and have high dielectric strength.

ii. **Semi-hermetic compressor**. This compressor is also known as accessible hermetic compressor or serviceable hermetic compressor. The unit is as shown in Figure 1.4. The main advantage of this compressor over the hermetic type is that its accessibility for repair during a compressor failure or for regular maintenance.

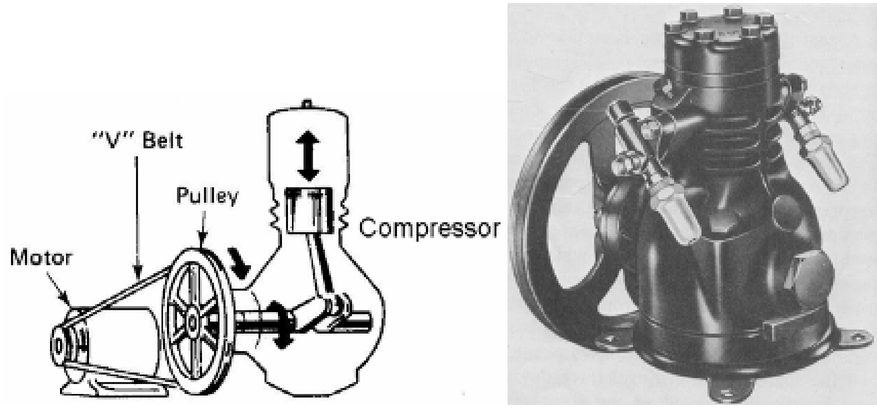
iii. **Open compressor**. In an open compressor, the compressor and the motor are enclosed in two separate housing as shown in Figure 1.5. This compressor needs the shaft seals to prevent refrigerant leakage. In most cases, an enclosed fan is used to cool the motor windings using ambient air. Notice that, there are two driving concept of the open compressor; belting drive and direct drive. An open compressor may be disassembled for service and preventive maintenance to the internal parts [3, 4, 5, 6].



A- Motor rotor B- Motor stator C- Cylinder
D- Piston
E- Connecting rod
F- Crankshaft G- Crankcase H- Shell
I- Electrical connection

Figure 1.3: Photograph of hermetic reciprocating compressor (Whitman and Johnson, 1991)

Figure 1.4: Photograph of semi-hermetic compressor (Whitman and Johnson, 1995)



(a) Complete assembly of compressor (b) Photograph of open compressor

Figure 1.5: Open compressor assembly type (Langley, 1982)

1.3 Research Overview

The application of reciprocating compressor in a refrigerator and an air- conditioner is already established. Chillers for some big building air-conditioning system are using screw compressors, but the researches are still on-going to improve the performance. Whereas automotive air-conditioning system are using both rotary and reciprocating types and again research in this area is actively pursued. Domestic refrigerator has been known to use reciprocating compressor until lately when rotary compressor has been introduced and appears to be successful. This success is as a result of continuous research carried out by the industry to improve the efficiency
and reliability of rotary compressors. As described later this compressor can be of static or rotating vane types.

The literature study has been done on the rotary and reciprocating compressors and findings showed that the performance of rotary compressor is better than reciprocating compressor. Recently, Universiti Technologi Malaysia (UTM) has developed a new compression concept comprises of a rotating vane, a rotating sleeve and a rotor. This is a simple concept compared to the other rotary compressors available in the market today. Details of the new rotary compressor concept are shown in Figure 1.6.

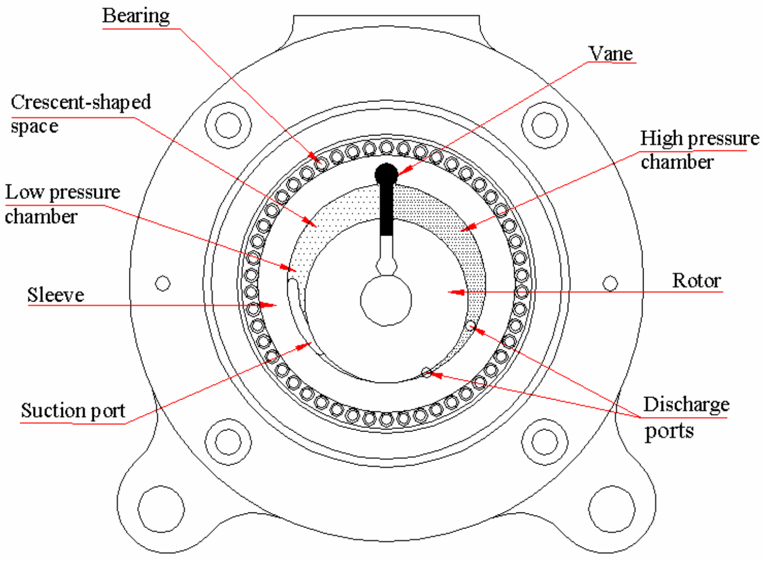


Figure 1.6: Concept of single vane rotating sleeve rotary compressor

The concept is equipped with main components such as a vane, a sleeve, a rotor, a shaft and a cylinder block. The sleeve as well as the cylinder block are assembled eccentrically to the rotor to produce the compression chamber with a crescent-shaped area formation. Then, the head or tip of the vane is assembled permanently into the slot on the inner part of the sleeve, and the vane hub is put inside the rotor slot. There is a contact point between rotor and sleeve to prevent the gas from leaking to the adjacent area. Therefore, the center points of sleeve rotation and rotor are different, and when the rotor starts to rotate, the vane will start to compress the gas. During first half of rotation, the sleeve pulls the vane out of the slot and during the second half the vane is pushed back into the slot, in the rotor. This concept is expected to reduce leakage through the vane tip which occurs in existing rotary compressor.

Figure 1.7 describes the operation sequence of this concept. The sequence starts at 0º with the compression chamber fully filled by gas and the high-pressure gas is completely delivered at 360º. The rounded vane tip allows it to swing so that kinematically the rotating mechanism works successfully inspire of eccentricity. The designed compressor will be installed to the household refrigerator-freezer. Generally there are three types of household refrigeration system, which are refrigerator unit, freezer unit and combination of refrigerator and freezer. This research is focused on combination of refrigerator and freezer unit. The combined unit is normally known as refrigerator [7].

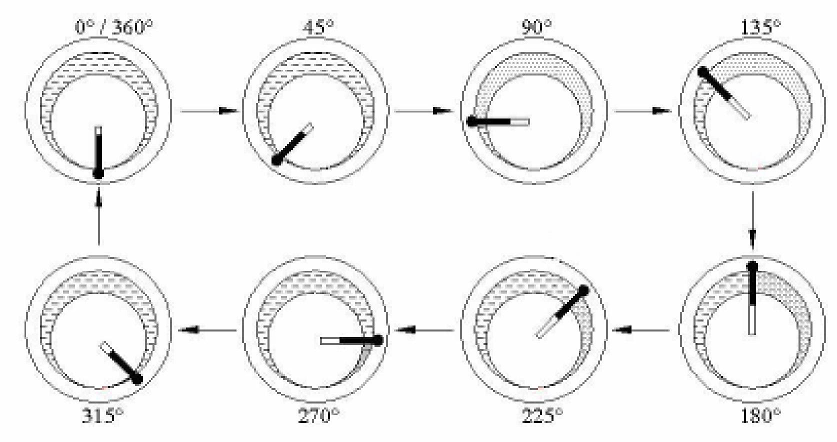


Figure 1.7: Operation of single vane rotating sleeve rotary compressor concept

1.4 Problem Statement

In Malaysia, the main refrigerator-freezer manufacturer is Matsushita Group of Companies with their products of National and Panasonic brands. Beside that, these companies also produce the compressor to be supplied to other refrigerator-freezer manufacturers such as Sanyo, Hitachi and Pensonic. Most of the refrigerator- freezers use reciprocating type compressor. However, there is also rotary type used in refrigerator-freezer such as for three doors refrigerator-freezer model. The presence of these companies in Malaysia gives us the opportunity to learn the technology and the manufacturing processes involved. This will gradually increase our capability in this industry and ensures that Malaysia remains the world biggest supplier of domestic refrigerators and split-unit air-conditioners. In this respect compressor is the most important component. Therefore, the main objective of this research is to develop our own technology of compressor. The research work is focus on the weaknesses of the two existing compressor models; rolling piston and sliding vane rotary compressor. As a result, a superior version of compressor will come out. The weaknesses on the existing compressors are leakage and friction problem through the vanes tip and cylinder block or rolling piston during compression process which will be discussed later.

1.5 Significant of Research

The sliding vane rotary compressor has a fairly high volumetric efficiency. The rolling piston rotary compressor has even higher efficiency. However from the literature review and theoretical analysis conducted the performance of rotary compressor can be further improved and manufacturing cost reduced. This is indeed a good starting and to proceed to a more interesting and useful R&D work in the effort to fully acquire this very important technology.

1.6 Objective of Research

The main objective of this study is to design and develop a rotary compressor based on a new concept, which is a single vane rotating sleeve rotary compressor for refrigerator application.

1.7 Scope of Research

The scopes of this research are described as follow:

1) Literature study
   This involves patent study, technical review and reverse engineering work. The outcome from this study will be adopted into the new compressor design.

2) Concept and design development

Design a new rotary compressor based on the rotating vane and sleeve concept.

CHAPTER 2

DESCRIPTIONS AND REVIEWS OF REFRIGERANT COMPRESSORS

2.1 Introduction

This chapter discusses relevant reports on the description of various concepts of rotary compressors and a description on the concept of reciprocating compressor that is installed in domestic refrigerators. Reviews are made on the continuous development of the rotary compressors from the documents of the relevant patents and technical reports available in the literature. Finally the chapter reports on the result of the reverse engineering work carried out on the existing rotary and reciprocating compressor models.

2.2 Descriptions of Compressors

2.2.1 Rotary Compressor

Rotary compressor is a machine which compresses the gas as a result of the angular movement of the vane or roller. Rotary compressors are normally suitable for application of low compression ratios and for small and medium gas flowrates. Rotary compressors have certain advantages such as continuously flow process, high speed of rotation and the design can be scaled down to a vary small dimension. The design of this compressor does not require suction valve and installation of discharge valve is optional. No clearance volume is required. Rotary compressor, on the other hand, has well known disadvantages that each of them requires high precision in machining, correct tolerance to balance between internal leakage and friction and requires continuous cooling to prevent from mechanical jamming [8].

In its applications, the rotary compressors are divided into four types; rotary sliding vane compressor, rolling piston rotary compressor, scroll compressors and gear compressor. However, the discussion in this chapter will be focused only to the sliding vane and rolling piston rotary compressors.

2.2.1.1 Sliding Vane Rotary Compressor

The sliding vane rotary compressor employs a series of rotating vanes or blades that are installed equiangle around the periphery of a slotted rotor. This machine has several components that are assembled into the housing [9]. Figure 2.1 shows the main components of the compressor and the arrangement within the housing or casing.

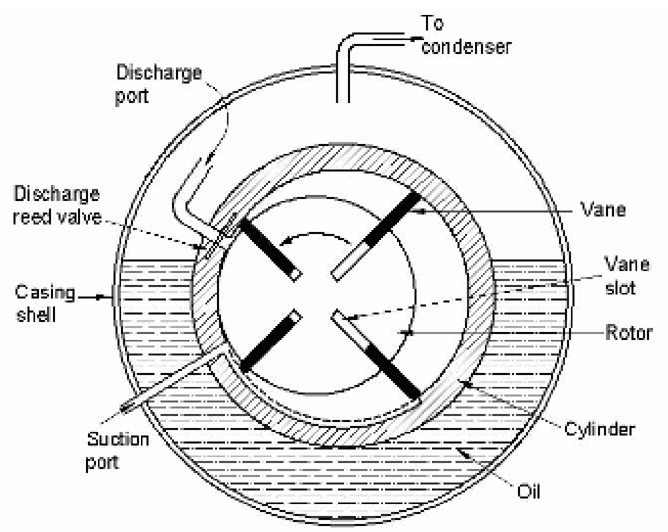


Figure 2.1: Illustration of sliding vane rotary compressor with 4 vanes

Basically, this concept is easy to understand and the operation only involves the vanes and the rotor. The rotor is mounted eccentrically in a steel cylinder to create the crescent shape compression chamber and the rotor is barely touching the cylinder wall at one point at which the two surfaces are hydraulically separated by oil. Deep slots are milled into the rotor, mostly radial but sometime oblique as shown in Figure 2.2 [8].

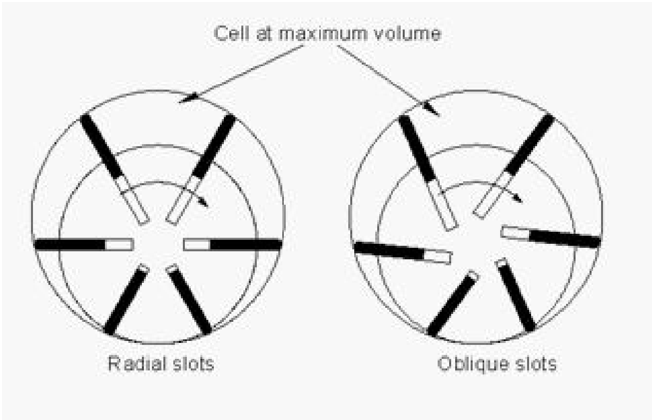


Figure 2.2: Slots arrangements in rotor

As mentioned earlier, the arrangement of the rotor, vanes and cylinder creates the crescent-shaped space between rotor and cylinder. This space is subdivided by the vanes into a number of compartments or cells. Directly opposite to the contact point between the rotor and cylinder wall the volume of the cell is maximum at which suction ends and compression begins. When the rotor starts to rotate, centrifugal force pulls each vane out until the tip touches the cylinder wall. Initially the cell increases its volume creating a vacuum which draws in the refrigerant gas into the low pressure cell through the suction port. As soon as the cell reaches its maximum volume suction ends and compression begins as rotation continues. The compressed vapour is discharged from the cylinder through a port located in the cylinder wall near the contact point between rotor and cylinder wall. The discharge port is so located as to allow discharge of the compressed vapour at the desired pressure [8, 9].

2.2.1.2 Single and Multivane Rotary Compressor

Single vane rotary compressor is usually used in a refrigerator or a small air conditioning unit where in both applications the cooling capacity is small. The weakness of a single vane rotary compressor is that as the compression pressure increases the internal leakage of the gas through the rotor and cylinder wall contact point and through the sides and tip of the vane, increases. The leakage becomes maximum when the pressure reaches its discharge point. As a result, the volumetric efficiency obtained is relatively low. The swept volume of single vane compressor equals that given by the crescent shape area.

The multivane rotary compressors are widely used in automobile air conditioning system. This compressor provides a double advantage. One is that there is a pressure differential across the crescent area resulting in a reduce pressure difference between adjacent cell. This result in a lesser internal gas leakage and therefore giving a higher volumetric efficiency. The gas in a compression cell may reach its discharge pressure well before the cell reaching the rotor/cylinder contact point. As such at this point there is not much gas left and leakage through the hydraulic gap is no longer significant. In addition the total swept volume of the compressor equals the maximum cell volume times the number of vanes. Therefore, the swept volume of compressor is depending on the number of vanes, where the large number of vanes can produce large swept volume. This virtually gives a bigger swept volume as compared to that of the single vane rotary compressor. All the above discussions are illustrated in Figure 2.3. It is not unusual to find a rotary compressor with 20 to 30 number of vanes depending on the application [8].

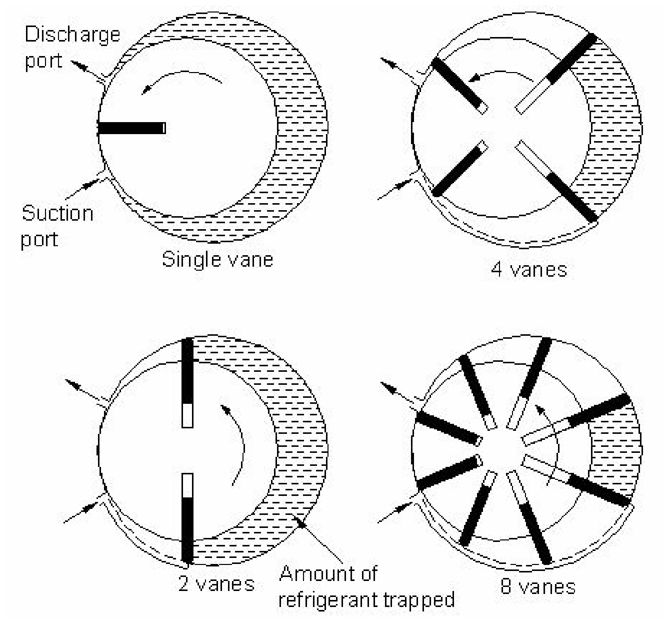


Figure 2.3: Diagram of single and multivane rotary compressor

2.2.1.3 Rolling Piston Rotary Compressor

Rolling piston rotary compressors, also called as fixed vane rotary compressors, are used in household refrigerators and air-conditioning units of capacity up to about 2 kW. The rolling piston rotary compressors employ a cylindrical steel roller which revolves on an eccentric shaft, the latter being mounted concentrically in a cylinder as shown in figure 2.4 [9, 10].

The compressor consists of several main components; which are a concentric shaft, a roller piston, a vane and a cylinder block. The arrangement of the roller piston is eccentrically to the cylinder and touches the cylinder wall at the point of minimum clearance. This compressor has a single vane which slides in and out of external slot located in a cylinder block [11]. At the base of the slot is a spring which pushes the vane out so that the tip is always in contact with the rolling piston, thus separating the high and low pressure chambers. End plates are used to close up the cylinder at each end and to serve as supports for the camshaft. Suction and discharge ports are located in the cylinder wall near the vane slot, but on the high pressure side. The vapour flows through the suction port and is continuous because no valve is required and the reed valve is assembled at the discharge port. The suction and discharge vapour is separated in the cylinder at the point of contact between the vane tip and roller on one side and between the roller and cylinder wall on the other side [9].

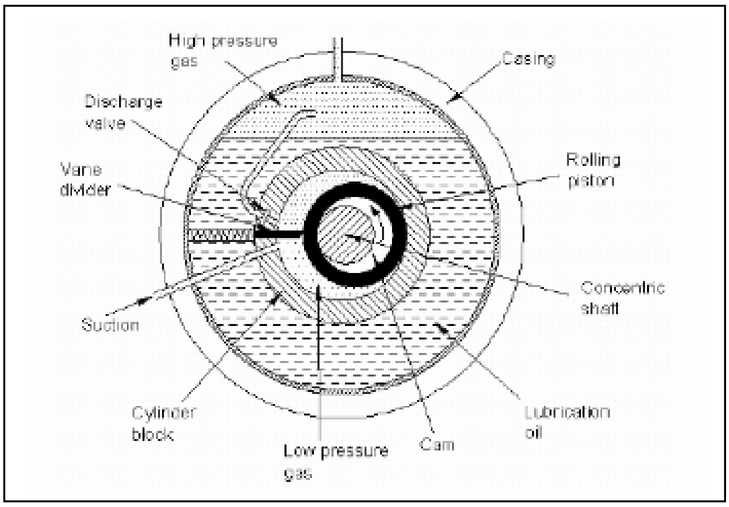


Figure 2.4: Illustration of rolling piston rotary compressor

The sequence of the compressor operation is illustrated in Figure 2.5 [12]. Initially, the vapour is induced through a suction port to fill the compression chamber which is closed by a vane and the contact line such as illustrated in Figure 2.5(a). As the shaft starts to rotate, the rolling piston rolls around the cylinder wall in the direction of shaft rotation and always in contact with the cylinder wall. When the rotor has rolled over the suction port, the vapour in the cylinder is steadily compressed. The discharge process will occur when the compressed vapour is slightly higher than that of the vapour at outside of compression chamber resulting the discharge valve to open and the compressed vapor will be pushed out to the discharge line.

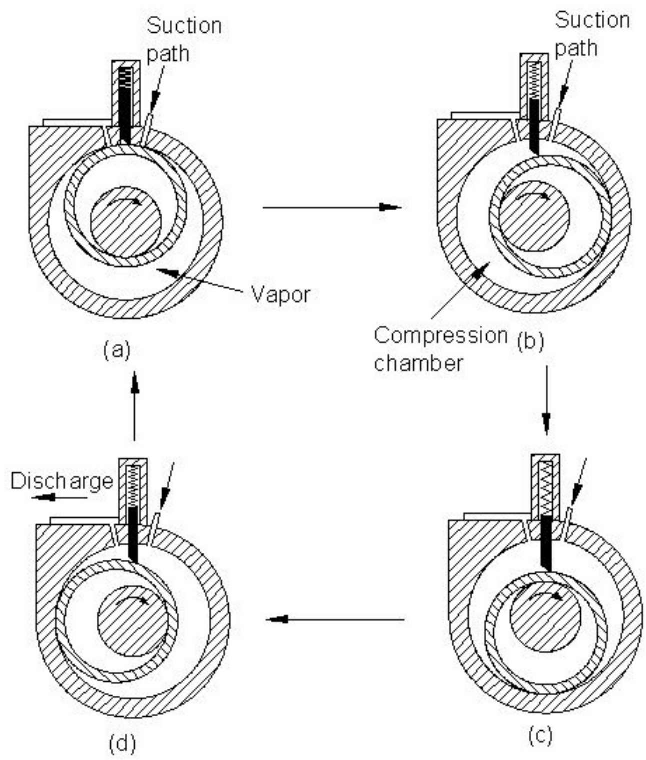


Figure 2.5: Operation principle of rolling piston rotary compressor

The rolling piston rotary compressors have a very small clearance volume same as sliding vane compressors. As a result, these compressors can produce higher volumetric efficiency compared to that produced by a reciprocating compressor. The volumetric efficiency also depends on the internal leakages during compression. The internal leakages can be minimized through hydrodynamic sealing, mating parts selection and clearances application. The hydrodynamic sealing is created by an oil film formation on a pair of rubbing surfaces and depends on clearance, surface speed, surface finish and oil viscosity. The high mechanical efficiency is achieved by minimizing friction losses that occur between the vane and slot wall, vane tip and roller and between cam, roller and end plates. The rolling piston compressor also produces less noise and vibration as a result of the installation of weights that counter-balance the rolling piston during the operation [9].

2.2.2 Reciprocating Compressor

Reciprocating compressor derives its name from the backward and forward movements of the piston to respectively induce and compress the gas in the cylinder [13]. The compressor components such as piston, crankshaft, connecting rod and cylinder are similar in design to that of the internal combustion engine [14]. These components have been arranged in the crankcase to produce the reciprocating motion. Figure 1.3 shows a typical internal assembly of a hermetic reciprocating compressor.

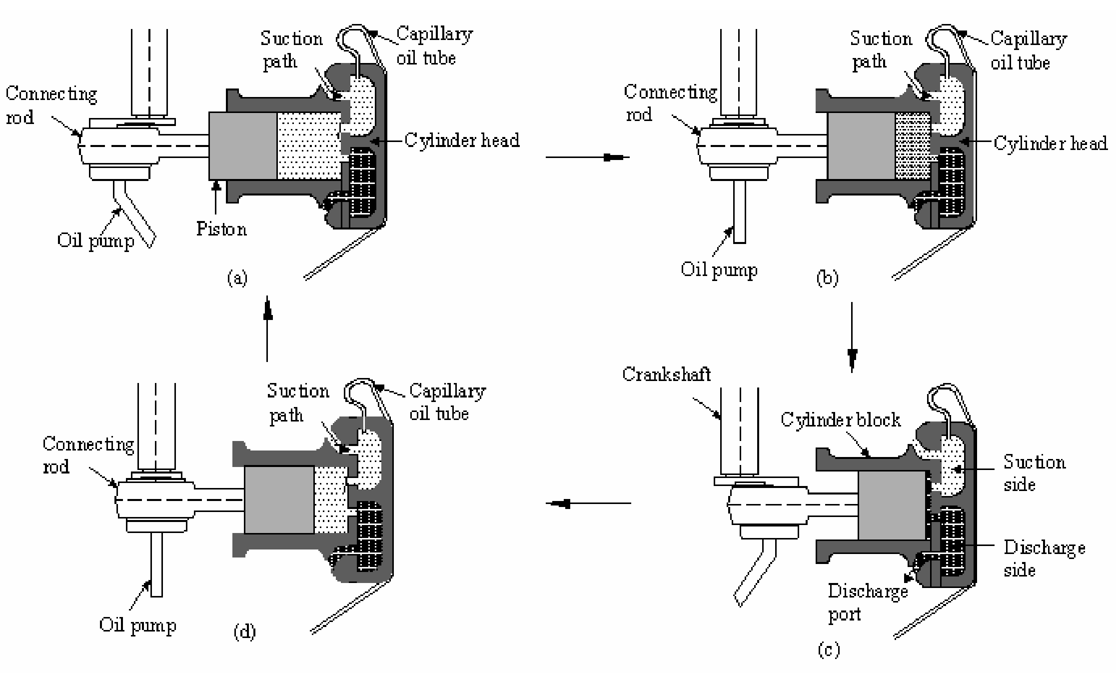


Figure 2.6: Operation principle of a reciprocating compressor

Figure 2.6 shows the operating principle of the reciprocating compressor. As the piston moves downward (or to the left) a vacuum is created and the suction valve opens and low pressure refrigerant vapour from evaporator fills up the cylinder until the piston reaches the bottom dead center (BDC) as shown in Figure 2.6 (a). Figure 2.6 (b) shows the compression stroke. At first both suction and discharge valves are closed as the pressure increases and volume decreases. At pressure which is slightly higher than that of the condenser the discharge valve opens and the gas is released and flows into the condenser as shown in Figure 2.6 (c). To allow movement of suction valve the piston stops at the top dead center creating the space called a clearance volume which is occupied by the residual high pressure gas. The residue will expand and mix with fresh gas during suction stroke to complete the cycle as shown in Figure 2.6 (d) [15, 16].

2.3 Reviews

Compressors of sliding single or multi-vane, single vane rolling piston and finally of reciprocating types have been respectively described. These three types are that normally used for low thermal load application like in a domestic refrigerator, in a small split unit room air conditioner or in a small automotive air-conditioning system. These types are further studied by reviewing:

i) The relevant patents,
ii) The literature on technical papers from proceedings and journals, and
iii) The designs of existing models that are available in the market.

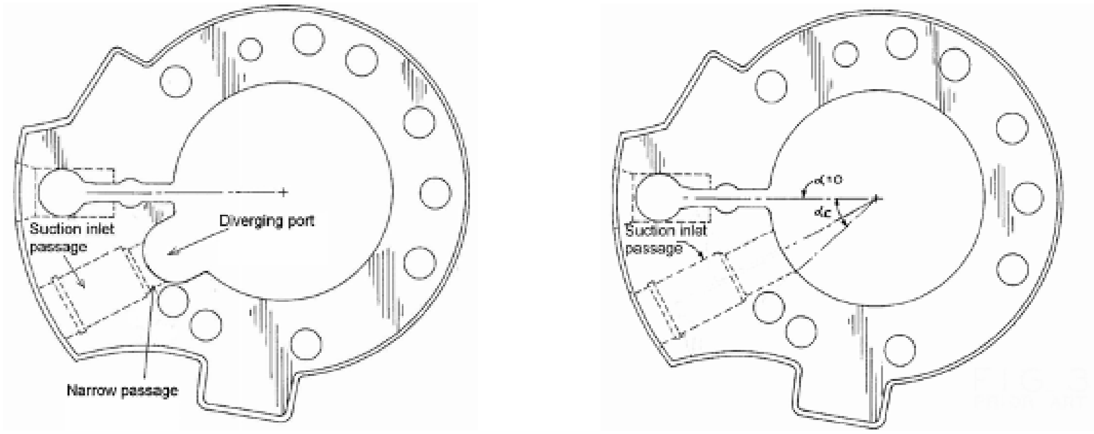
2.3.1 Patents Review

There are thousand of patents on compressor in US patent file alone. This file is dated as early as 1873. However, there are probably about half are on rotary types which were filed as early as 1911 followed by innovative improvement efforts showing a trend that it will continue into the 21st century. The reciprocating type is an old and already established technology and lately very few and minor innovative efforts are filed and successfully patented. As such the patent review carried out in this work is focused only on rolling piston and sliding vane rotary compressor types.

2.3.1.1 Patent of Rolling Piston Rotary Compressor

The rolling piston concept was introduced earlier in 1911 by Kinney, J. R. [17] who developed the rotary pump. In 1933, Buchanan, J. C. and Hubacker, E. F. [18] introduced an improvement in the discharge system by using a flapper valve. Warrick, L. K. *et al*. [19] improved the rolling piston concept design for refrigeration compressor. They developed the seal housing assembly method for compressor and motor, and improved the lubrication arrangement mechanism. Since the compressor was introduced in last several decades, there are several modifications in design to increase volumetric efficiency, reduce noise, improve balancing, improve sealing and etc.

Dreiman N. I. [20] introduced an improvement on the suction inlet for rolling piston rotary compressor. The suction inlet passage was designed with an entrance passage, a narrower passage and a divergent port into the compression chamber. The suction inlet passage serves as a diffuser while the narrow passage functions as a throat to the diffuser to increase the volumetric efficiency with respect to the suction gas entering the cylinder. This improvement increased the volumetric efficiency, reduced pulsation and the associated noise and increased the pressure of the suction gas in the cylinder at the beginning of the compression cycle. Details of the design is shown in Figure 2.7.



Innovation by Dreiman Prior design

Figure 2.7: Innovation of suction inlet rolling piston rotary compressor by Dreiman, N. I

Besides that, the discharge passage also had some improvement to increase the compressor performance. Costa, D. and Caio, M. F. N [21] introduced a discharge system through bearing plate which uses an orifice and supported by a valve plate. There are two orifices which are drilled on the bearing plate according to the area of the discharge volume. First orifice is drilled at about 330° on the bearing plate, whereas the second orifice is located at about 350° of the shaft rotation. Figure 2.8 shows the operation principle and locations of orifices on bearing plate corresponding to the compression mechanism in cylinder. At the beginning of discharge, both orifices open to allow the relatively large quantity of gas to flow out. When the volume of compression becomes smaller, the first orifice is closed by the rolling piston, and the gas is completely discharged through the second orifice.

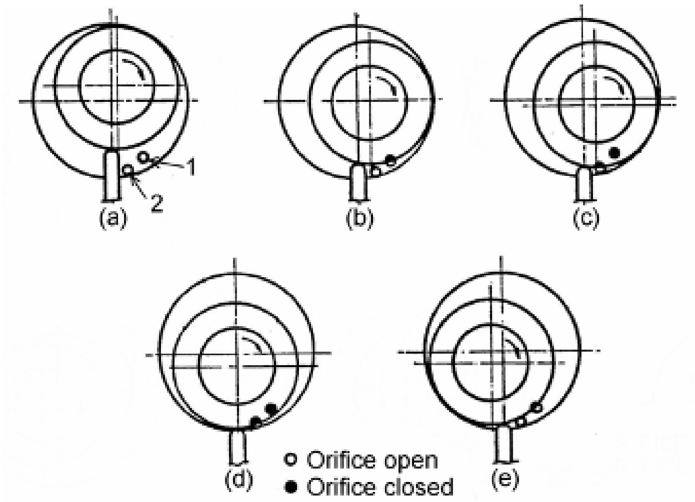


Figure 2.8: Sequence operation of two-orifice discharge method

Heui-Jong, K [22] introduced a new design of compression mechanism based on rolling piston rotary compressor as shown in Figure 2.9. The main objective of the inventor was to eliminate the piston vibration caused by difference of gas pressures and also to eliminate the use of valve to reduce noise. He designed a compressor which has a cylinder casing driven by an electric motor. The cylinder casing is equipped with two vanes that are controlled by a loop spring and a C shaped snap ring. The loop spring and C shaped snap ring will control reciprocally the vane movement. Cylinder casing is also equipped with two-discharge passages and closed by two end plates at the top and bottom respectively. The top end plate is very important whereby it is cut with grooves to form a suction port and discharge port respectively. There is an eccentric shaft at the bottom side of top plate to roll the piston continuously. Suction gas enters and fills the compression chamber and discharge occurs when the discharge passage meets discharge port at upper side plate. High-pressure gas completely discharged when the discharge passage cross over the discharge port. This cycle will occur twice per rotation. As a result this compressor is capable to deliver gas quantity more than that of conventional design. Figure 2.9 shows the design of this compressor and parts involved, whereas Figure 2.10 shows the operating principle of the compressor.

The compressor is designed to overcome the problems which occur in conventional rolling piston rotary compressor. The previous design produced unbalance force consequence of pressure difference between suction side and compression side, as a result vibration was generated during the compression process.

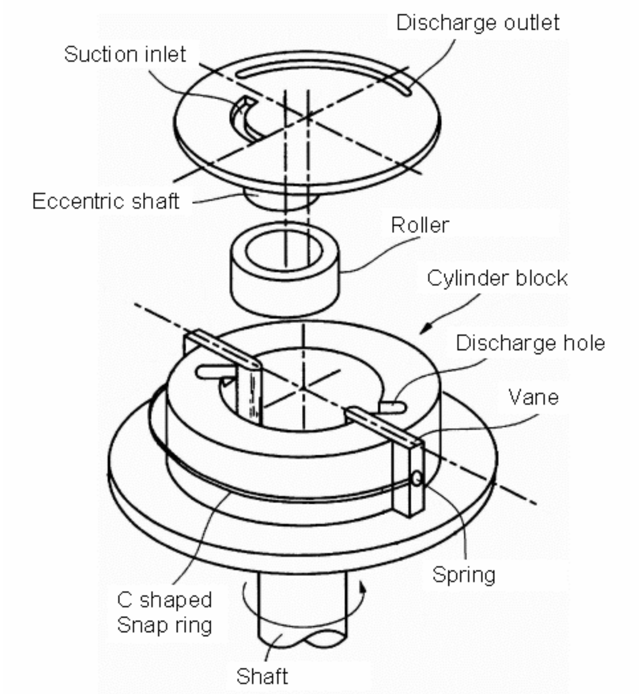


Figure 2.9: Dual chamber rolling piston rotary compressor

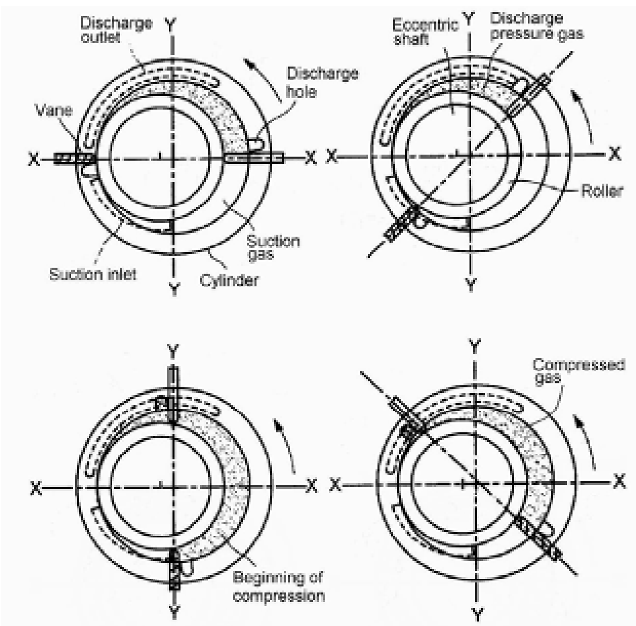


Figure 2.10: Operating principle of dual chamber rolling piston rotary compressor

2.3.1.2 Patent of Sliding Vane Rotary Compressor

The basic idea of sliding multi-vane rotary compressor design began with a single-vane concept. Since then, the improvement has been made to increase the gas quantity delivered, efficiency and performance of compressor by increasing the vane numbers and modifying the compression chamber shaped. Previously, Gillespie, J. E [23] introduced a single vane rotary pump with eccentric ring. The advantage of this pump is having two compression chambers. One chamber between rotor and eccentric ring, and other one is between eccentric ring and outer cylinder. Walter, J. P [24] introduced the multi-vane rotary pump for water. Camilo, V. N [25] introduced rotary vacuum and compressor pump. In this invention, he reduced frictional losses and wear, and also reduced contact pressure between vane tip and cylinder wall by designing the vane with light material. Otherwise, Adalbert, G., *et al.* [26] introduced a new sliding vane compressor completed with oil sealing at clearance gaps. The oil was supplied from the high-pressure tank of the compressor to the clearance gaps between the rotor end faces and lateral stationary faces of the housing.

The design of a sliding vane rotary compressor was further improved by introducing two oval shaped compression chambers. The rotor and cylinder block were assembled concentrically and having two contact points at opposite positions of each other. The contact points divide the cylinder bore into the two crescent-shaped spaces. The operating principle of this compressor is similar to that with a single chamber sliding vane rotary compressor. Adalbert, G and Vysiotis [27] designed the dual chamber rotary sliding vane compressor for a vehicle air conditioner. The advantages of dual chamber rotary compressor are providing more quantity gas delivered and reducing the friction loss to one half that for a single chamber rotary sliding vane compressor. Figure 2.11 shows the detail of the compressor.

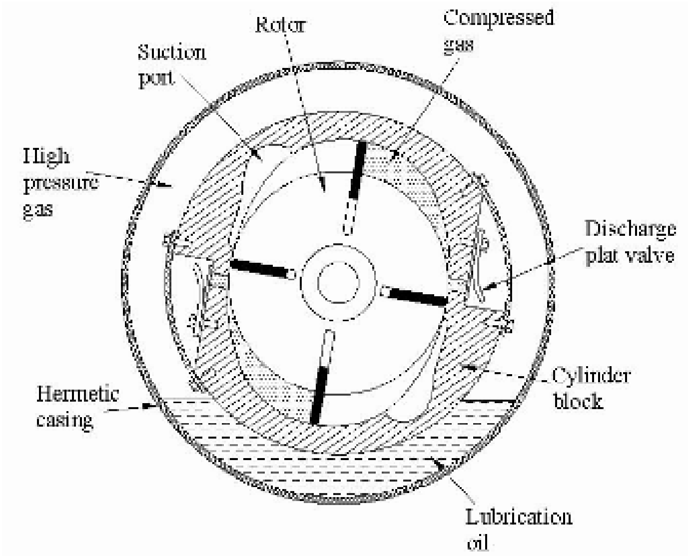
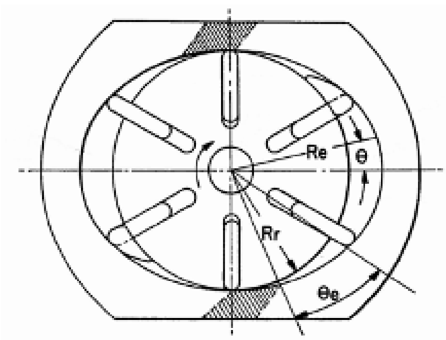


Figure 2.11: Dual chamber sliding vane rotary compressor patented by Adalbert and Visiotis

Figure 2.12 shows the configuration of dual chamber rotary sliding vane compressor which was patented by Cavalleri, R. J [28]. Figure 2.12 (a) shows a compressor that has elliptical contours which, the widening of the elliptical shape could provide the maximum suction intake. However, the elliptical contours were modified to provide a larger initial starting volume and a correspondingly smaller discharge volume as shown in Figure 2.12 (b).



(a) (b)

Figure 2.12: Configuration of dual chamber sliding vane rotary compressor patented by Cavalleri

2.3.2 Literature Review

2.3.2.1 Design Geometry

All of the compressors were designed with different of geometric parameter. In fact, the geometric parameter for same compressor type can be different depending on the size of the compressor, application and working pressure. In order to develop a good compressor design, the correct geometric parameter should be applied. Meece, W. *et. al* [29] discussed the technology to design oil-less reciprocating, rotary vane and diaphragm compressors and pump. The study was focused to the rotary vane compressor and pump. The characteristic of oil-less rotary compressor and pump is similar with current rotary vane compressor. The operating concept of oil-less rotary vane compressor is same with the oil flooded rotary vane compressor. The current design of oil-less rotary vane compressor presented is up to 50 SCFM and produces a vacuum of 26 in Hg or a pressure of 20 PSIG at operating speed from 500 to 3600 RPM. To design a good oil less rotary vane compressor, authors have applied a rotor housing ratio or sometime called as design ratio of 0.8 to 0.9. Where the lower ratio was applied to vacuum pumps and a compressor of pressure up to 15 PSIG and the higher ratio was applied for compressor of pressure up to 80 PSIG. Figure 2.13 shows the geometry of the compressor.

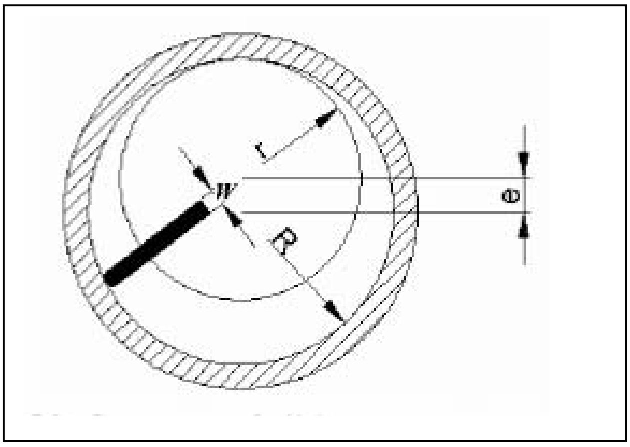


Figure 2.13: Geometry of oil-less rotary vane compressor

2.3.2.2 Performance

Performance of a rotary compressor depends on the control of key clearances between the respective moving parts. The gaps in these key clearances should receive adequate lubricantion oil to reduce friction, wear and leakage. The effect of leakage in a rotary compressor is to reduce the delivered flow of refrigerant, reducing the cooling capacity and volumetric efficiency, and increasing the compression power. Pandeya, P. and Soedel, W. [30] identified the various losses in a hermetic type rolling piston rotary compressor. These losses are divided into two broad categories; the energy losses and the mass flow losses. The energy losses involving motor loss, friction loss, compression loss, valve loss and lubricant pump loss. Whereas, the mass flow losses involving clearance volume loss, leakage loss, back-flow loss, suction-gas heating loss and loss due to lubricant flow. This paper only discusses two types of losses that occur due to friction in the cylinder and that due to leakage. In the investigation, there are 6 points of friction losses in cylinder and 4 points of leakage losses have been identified. They developed mathematical model for each losses point. As a result, the predicted ideal mass flow of this compressor is 8.84 kg/hr with the leakage loss of approximately 12% and cylinder friction loss approximately 9%. T. Matsuzaka and S. Nagatomo [31] discussed the performance analysis of rolling piston rotary compressor. They analyzed the losses factors that affect the compressor efficiency theoretically and experimentally. They optimized the discharge port shape corresponds to maximum compressor efficiency. As a result, the volumetric efficiency was increased to 94% and compression efficiency to 96%. Besides that, they have also improved the mechanical losses by considering the lubrication, reliability and vibration. As a result, they have increased the mechanical efficiency to 94%. Through this method, they found that the maximum compressor efficiency was 74%. H. Kawai [32] introduced a high efficiency horizontal rolling piston rotary compressor for household refrigerators. An experimental analysis was done to determine the performance characteristics of the compressor looking at three things:

1) Assumption of excessive compression loss by making the entire discharge system as one orifice.

2) Determination of a clearance between the blade and piston.

3) Efficiency improvement and noise reduction through design improvement of the bearing.

It was found that, an excessive compression in the compressor occurred due to the small size of discharge port designed. This was discovered after it was analyzed by simulating the discharge system as an orifice model. The consideration also made to the effective flow area of discharge port and experimental simulation was done to the various discharge port sizes to optimize the design of the discharge system. As a result, 3 to 4 % of efficiency improvement was realized. S. Nagatomo et al. [33] discussed the performance analysis in a rolling piston type hermetic compressor that was used in domestic refrigerators. They analyzed the two main influencing factors on the compressor efficiency. Those are roller clearance and the discharge port length. In order to measure the performance of the compressor, the test model was developed by assembling three units of Piezo type pressure transducer at locations of □□31□, □□230□ and □□353□. An eddy current probe was mounted on the center of the valve stopper to measure the discharge valve behavior. The roller clearance was varied and when the roller clearance was large, the volumetric efficiency was reduced and compressor intake power increased. This was caused by the oil leakage to the suction side of the cylinder from the inside of the roller through the axial clearance. Four kinds of discharge port lengths were used in this experiment. The length of discharge port influences compressor efficiency by means of re-expansion of the residual gas in the discharge port. If the discharge port length becomes large, the quantity of residual gas in the cylinder increased resulting in a decrease in compression efficiency. Result from the analysis of the various sizes of roller clearance and discharge port lengths show that the isentropic work improves from 48.2 % to 55.2 %. T. Nomura et al. [34] discussed the method to improve the efficiency of rolling piston rotary compressor. They also carried out an investigation to identify power losses in previous rotary compressors. It was founded that the overall efficiency is about 57.2 % and the largest power losses are the motor loss that accounts for about one half of the total power loss. From the losses analysis, the counter-measures were planned for reducing those losses. The conformation tests on the effectiveness of these measures were also conducted, than the counter-measures that have the large effect to the performance improvement were selected and were applied to real machine. As a result, the overall efficiency was increased by about 12 %. H. Shintaku et al. [35] introduced a swing piston type compressor based on rolling piston rotary compressor. The designed compressor has reduced the sliding loss and contact pressure between roller and vane tip. The compressor was designed with a groove machined on the outer surface of the conventional rolling piston so that the vane tip can be entered into the grove. The differences in operation of this compressor compared with the conventional rolling piston are; (1) the swing movement of the piston and (2) the swinging piston structure has a groove with a cross section of a circular arc and a radius almost same as the vane tip radius. This is to provide swinging motion and to secure the contact area between vane tip and piston so that it can reduce the contact pressure and improve the sealing performance. The feasibility of this compressor was examined both theoretically and experimentally. As a result, the mechanical efficiency is increased about 1 % and COP about 1 % based on theoretical analysis. Experimental result had shown that the improvement in mechanical efficiency is about 1 %, COP is about 2.2 %, indicated efficiency is about 1.1 %, the increment of volumetric efficiency is about 0.7 % and compressor efficiency is about 2.9 %.

Y. Huang et al. [36] introduced a new two stage rolling piston rotary compressor design for automotive air conditioning system. The objective is to achieve a well-balanced performance, Noise-Vibration-Harshness (NVH), durability, variable capacity, manufacturability and cost of operation. The compression principle of this compressor is shown in Figure 2.14. Low-pressure refrigerant vapour is introduced into the first stage compression chamber via ports situated in the sliding vane. First stage of compressed gas is discharged into an intermediate plenum in the rear housing through a reed valve. Then the intermediate refrigerant gas is introduced into the second stage chamber formed by inner wall of rotor and the outer wall of the post to complete the full compression.

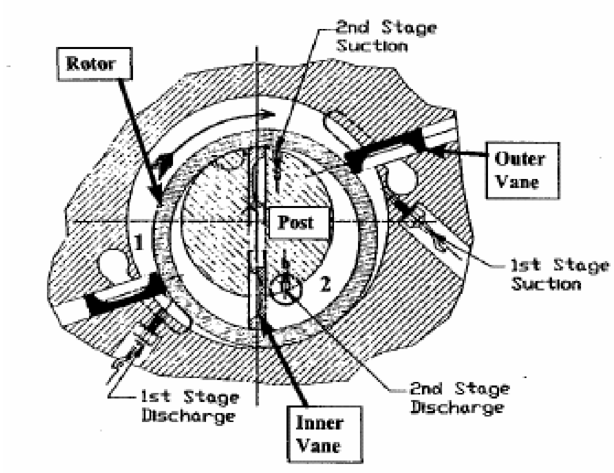


Figure 2.14: Compression principle of advanced rolling piston rotary compressor

This compressor applied a Taguchi Method [37] in the design of experiment (DOE) and to optimize the design significant parameters that affect the compressor performance were studied. The optimization gave the minimum possible power in order to achieve the desired cooling capacity. As a result, they obtained a COP of 1.56 and volumetric efficiency of 81 % when operating at 1000 rpm. At 3000 rpm, the COP was 1.32 and the volumetric efficiency was 76 %. The same compressor concept was applied for semi trailer or truck [38] and was tested more than 150,000 miles.

2.3.2.3 Leakage

Internal leakage is a normal phenomenon which occurs in a compressor. Internal leakage influences the performance of the compressor. In a rolling piston rotary compressor the major leakage occurs at the clearance exists between the external surface of the rolling piston and the internal surface of the cylinder, known as radial clearance. The effect of leakage in a rotary compressor is to reduce the delivery flow of refrigerant, reducing the cooling capacity and volumetric efficiency, and increasing the compression power. Reed, W. A. and Hamilton, J. F [39] discussed the internal leakages of a sliding vane rotary compressor. They identified five internal leakage paths and developed mathematical model to predict the instantaneous internal leakage magnitudes. The five internal leakage paths are depicted in Figure 2.15. To simplify analysis, the compressor is divided into three-control volumes which are a suction chamber, an intermediate transfer chamber and a discharge chamber. In the analysis of the leakage paths through the minimum clearance, past the blade edge, and past the blade tip, two extreme cases have been analyzed to establish upper and lower bounds for the leakage magnitude. Upper bounds were established by assuming the leakage fluids to be entirely compressible refrigerant gas completely filling the clearance space. Lower bounds were established by assuming the leakage fluids filling the clearance space to be an incompressible mixture of lubrication oil diluted with an equilibrium concentration of refrigerant gas. Figure 2.16 summarizes the analysis of the individual internal leakage magnitudes by rating the various leakages. The most significant internal leakages were found occurs through the lubricating oil system (iv) and through the minimum clearance

(i). Secondary leakages are past the blade edges
(ii) and from the discharge valve transfer slot
(iii). Leakage past the blade tip
(v) was found to be negligible.

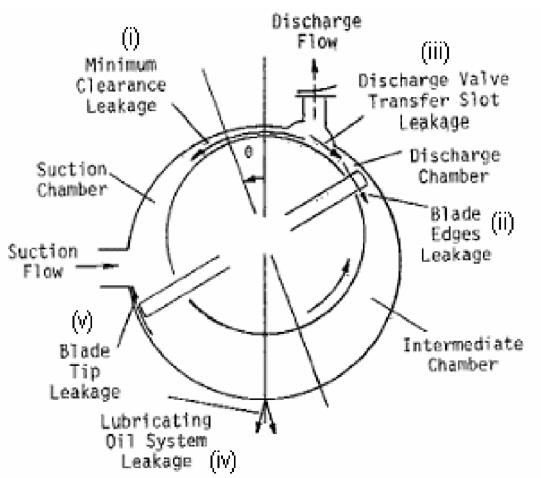


Figure 2.15: Definition points of internal leakage paths by Reed and Hamilton

Table 2.1: Comparison of leakage flow models

Leakage Component	Leakage Flow Fraction	
	Liquid Model	Gaseous Model
Into Suction Volume	0.094	0.485
Into Compression Volume	-0.004	0.092
Total Net	0.090	0.577

From the results, the liquid leakage model is most realistic than gaseous leakage model. The results also illustrate the need to maintain oil in the clearance gap of the leakage path to provide good sealing. There is not much difference between predicted value and measured value. As a conclusion, the clearance gap and oil sealing should be controlled in order to produce a good compressor performance.

Gasche, J. L et al. [41] introduced a model to calculate the transient flow of the lubricant oil through the radial clearance of rolling piston compressors by considering he time variation of both the pressure difference between the compression and the suction chambers, and tangential velocity of the rolling piston during its complete revolution. They developed the mathematical model to calculate the mass flow of oil and refrigerant using momentum and continuity equations. The analysis was done with various tolerances and oil temperature. It was founded that the oil temperature influenced the flow for various minimum clearances. The results show that the total oil leakage increases with both minimal clearance and temperature. The total gas leakage reduces with increasing temperature due to the solubility reduction of the refrigerant in the oil. Table 2.2 shows the results of analysis with various minimum clearances and oil temperature.

Table 2.2: Total refrigerant gas leakage through the radial tolerance

Tolerance □(μm)	Toil (°C)/ mass leakage in relation to the R22 being compressed (%)					
	80°C	%	100°C	%	120°C	%
10	0.00032	0.14	0.0003	0.13	0.0002	0.09
20	0.0017	0.72	0.0016	0.70	0.0012	0.51
40	0.0091	4.0	0.0089	3.8	0.0065	2.8
60	0.0249	10.8	0.0243	10.6	0.0179	7.8
80	0.0509	22.1	0.0497	21.6	0.0367	16.0
100	0.0887	38.6	0.0867	37.7	0.0640	27.8

2.3.2.4 Material Application

Material application plays an important role in a compressor performance. The material that is applied as compressor component must be able to support wear resistance so that the compressor can function smoothly during operation. Besides that, the material must also have good thermal properties to support the operation at high temperatures. Meece, W et al. [29] described the material characteristic for compressor components in order to design an oil-less rotary vane compressor. They were proposing that the rotor is preferably machined from a solid piece of cast iron and treated to improve wear resistance. For certain applications the rotor is machined from carbon materials. The machining of the rotor has to be held within close tolerances. A rotor surface finishing of 0.813 mm is desirable except for the finish in the rotor slots that requires 0.406 mm. The rotor slot edges are machined with a radius. The trailing side of the rotor slot is perfectly blended with the edge radiused to achieve a polished finish in the transition area. For housing material, cast iron is widely used. However, sintered metal materials are successfully being applied to smaller size. The pump housing is designed to assist in cooling since it represents the major component for convenient heat removal created by the pump. The inner surface of the housing should be machined to a minimum of 0.406 mm. This surface can also be improved for the benefit of vane life, leakage and smooth operation by hard-coating, hard metal spraying, induction hardening as well as treated by a hot quenching method. In an oil-less rotary compressor, a carbon composition vane material has been successfully used. The outstanding advantages of carbon vanes over other materials are the self-lubricant properties and low specific gravity. The low expansion rate of carbon material contributes to dimensional stability, which is a remarkable advantage. The low coefficient of thermal expansion makes it possible to maintain desired clearance between vane end plate and vane rotor slot. The vane has to be machined to precise dimensions. Gap between vane and end plate should be 0.0127 to 0.0381 mm depending on length and vane thickness, in order to obtain minimum leakage. The leading edge of the vane should be radiused to prevent chipping and to reduce carbon dusting. Komatsubara, T et al. [42] studied the material selection for a rotary compressor. In their study, they investigated the use of metal matrix composite (MMC) as a high strength and reliable lightweight material and the use of fiber reinforced aluminium (FR- Al) alloy for moving parts. The authors developed the fiber reinforcement Aluminium Alloys material to be used in the compressor. There are two methods introduced by the authors to produce the FR-Al alloys; powder metallurgical process and the squeeze casting process. New FR-Al alloys were used as the material for the vane and rotor. As a result, it passed all the performance tests on durability, light weight, excellent wear resistance, high strength and high chemical stability.

2.3.3 Design Review

In this section the review were carried out on existing models of reciprocating compressor, rolling piston rotary compressor and rotary sliding vane compressor. Reciprocating compressor is relevant because it is used in existing refrigerator models and a direct comparison on performance can be made with the proposed rotary compressor. The rolling piston rotary compressor and a rotary sliding vane compressor are used as a design reference.

2.3.3.1 Reciprocating Compressor Design Review

The refrigerator that was used in the study as the experimental rig has the reciprocating compressor with the following information or specification:

Manufacturer : Matsushita Compressor (M). Model: DA66C12RAY5
Refrigerant Charge : 160g / HFC 134a
Power Source : AC 240V / 50 Hz
Current : 0.56 amp.
Input : 116 W Displacement: 6.6 cm3
Speed : 2945 rpm
Pole : 2 pole electrical motor
Compressor Type : 1 cylinder single stage reciprocating.

A used compressor of the same model was dismantled and studied. It is found that this compressor has a spring mounted to absorb vibration caused by unbalance force that is generated during the operation. The bottom side of this compressor acts as an oil sump whereby this oil is used to lubricate the internal moving parts. The refrigerant enters the compressor through a suction tube, and fills the whole hermetic casing across the motor winding before entering the compression chamber through the suction passage. In doing so the oil helps to remove some of the heat from the motor winding and also helps to evaporate any liquid refrigerant that may enter the compression chamber. This compressor has suction and discharge mufflers to stabilize the refrigerant flow and to absorb the noise and vibration caused by the discharge process. Appendix A1 shows the cut-away view of the hermetic reciprocating compressor and Appendix A2 shows the exploded view with a detail function of each component.

2.3.3.2 Rolling Piston Rotary Compressor Design Review

The specification of rolling piston rotary compressor that is being investigated in this study is give below followed by its general description.

Manufacturer : Matsushita Electric Industrial Co., Ltd. Malaysia
Model : 2KS34OD3AAO1
Output Power : 1600 Watt (2hp)
Refrigeration Capacity: 5105 Kcal/h

Swept Volume: 34.1 cc

Refrigerant : R22

Lubrication Oil: Suniso 3GS Power Supply: 220/240 V, 50 Hz

Actually, the rolling piston rotary compressors are widely used in split unit air conditioners. There are several sizes of the compressor ranging from ½ hp to 2 hp with similar design configuration. This compressor can be divided into 4 compartments, which are the accumulator, the oil sump, compression mechanism and discharge compartment. The suction gas from accumulator directly enters the compression chamber through a suction tube and a suction port. The suction port is constructed radially on cylinder block to allow the refrigerant to enter the compression chamber continuously. The gas is compressed to the high pressure and discharged through the upper bearing plate. The discharged gas is accumulated in the upper compartment of hermetic casing that comprises of electrical motor to drive the compressor. At the lower compartment of hermetic casing is assembled with oil sump to lubricate the entire moving parts. The lubrication oil is supplied to the compression mechanism through the pump that is located at the bottom of the shaft. Appendix B1 and Appendix B2 respectively show a section view and component boundaries of the compressor. The important finding of this investigation is the valve application, the tolerance applied and the material used. The discharge concept of this compressor is using a reed valve. The discharge will occur when the pressure in the compression chamber is slightly higher than the pressure in the discharge tank. There are six main components assembled to form this compressor; an eccentric shaft, a roller, a cylinder block, a vane, an upper bearing plate and a lower bearing plate. Most of these components are made from cast iron with hardening process except the vane and the eccentric shaft. The vane material remains proprietary and no attempt was made to conduct any analysis. However, Matsumoto, K et al. [43] in their work introduced a vane material made of stainless steel or tool steel with nitriding surface treatment, whereas, the shaft material is carbon based, manufactured by using powder metallurgy process. Important dimensions are taken including tolerances which are listed as follow:

1) Tolerance between cylinder block surface and outer peripheral of roller is about

3 to 5 µm.

2) Tolerance between moving component (such as vane, roller, and eccentric cam) and bearing plate is about 5 to 8 μm.

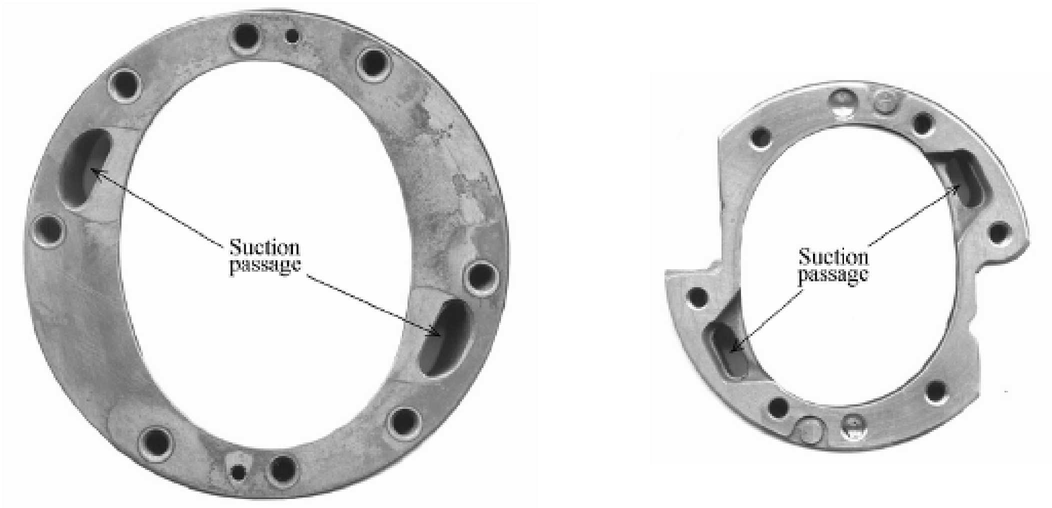
2.3.3.3 Review of Sliding Vane Rotary Compressor

Sliding vane rotary compressors are widely used in automotive air conditioner. There are two types of sliding vane rotary compressor investigated in this study. One is made by Denso for Perodua Kancil car and the other is by Patco for Proton Wira car. The characteristics of these compressors are shown in Table 2.3.

Table 2.3: Characteristics of Denso and Patco compressor

Descriptions	Denso for Kancil	Patco for Wira
Model	MA447220-6100	4G1AT-17064
Number of vanes	5	5
Ref. gas type	R-134a	R-134a
Displacement volume	72	144
Lubricant oil	MG-20	RL212B (Ester)
Low pressure test	1.67 Mpa	1.6 Mpa
High pressure test	3.53 Mpa	3.0 Mpa

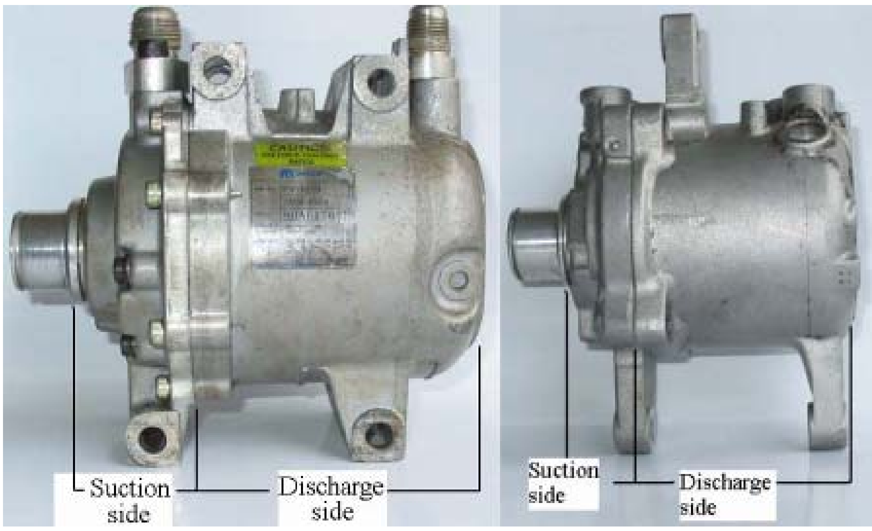
Both of the compressors are of dual chamber type sliding vane. The operation of these compressors is similar to the design that was developed by Cavalleri, R. J [28], whereby the suction gas enters from front cover and flow into the compression chamber through the front side plate. The suction path was designed on both sides of each chamber to ensure that enough quantity of gas fills the compression chambers at all speed. To accomplish this, the suction passages are made in the cylinder block right through to the rear side and the suction port sectors were machined on both sides to provide maximum gas entrance. Figure 2.18 shows the cylinder block of both compressors.



(a) Patco Compressor (b) Denso Compressor

Figure 2.18: Cylinder block for sliding vane rotary compressor

Generally, both compressors are divided into the two compartments: the suction and the discharge as indicated by Figure 2.19.



(a) Patco compressor (b) Denso compressor

Figure 2.19: Compartments boundaries of sliding vane rotary compressor

Low-pressure gas enters the compressor chamber through the suction passage and discharged through the discharge port at cylinder block closed to the sealing point. Each compressors uses discharge reed valves. In Denso compressor, there are three-discharge ports combined with three-reed valves. Whereas in Patco compressor, there are only two-discharge port combined with two-reed valves. An investigation to the discharge ports of both compressors was made. It was found that the discharge valve seats are different in the two compressors. In Denso compressor, it is just separated by groove between the discharge ports and bolt holes, whereas in Patco compressor, it has lip and groove at the discharge ports. Another important thing in these compressors is the material used. Most of parts in these compressors are using aluminium alloy material such as for rotor, sleeve, casing and vane. The surface of vane was treated by hard chrome plated and rotor surface is coated by carbon. For end plate, the material used is aluminium alloy A390 also with a carbon surface coating. Appendix C1 and C2 show the dismantled components of these compressors.

2.4 Conclusion

There are lots of useful information that was collected from the study that has been conducted. This information will be considered in order to produce a good design of a new rotary compressor such as design ratio, material selection, tolerance and surface finishing. As such in this design, the author has applied the design ratio proposed by Meece, W et al. [29]. Where, the proposed design ratio for sliding vane rotary compressor is between 0.8 and 0.9. This ratio is desired because it provides a wide of range to design the compression chamber. In addition, the new rotary compressor that will be designed is differing from existing rotary compressors. Based on the design review of the existing rotary compressors, Aluminium alloy A390 was selected to build-up first prototype of new rotary compressor. The tolerances of rotating components were referred to Gasche, J. L et al. [41] and measurements on rolling piston rotary compressor which has the maximum tolerance allowances as 10 µm. The surface finish of compression components should be equal to mirror surface to reduce friction between rubbing component.

CHAPTER 3

COMPRESSOR DESIGN AND DEVELOPMENT

3.1 Introduction

This chapter discusses the design and development of the new rotary compressor prototype in the aspect of the compression concept, design geometry, system design, material selection and quality of machining. The design of this compressor is based on the specification of existing reciprocating compressor which is used in refrigerator NR- B33TA National. The volume of the reciprocating compressor is 6.6 cm^3 and the operating speed is almost 3000 rpm.

3.2 Design Step of New Compressor

3.2.1 Design of Compression Concept

This compressor is called Single Vane Rotating Sleeve Rotary Compressor because both the vane and the sleeve rotate together with the rotor as described in 1.4. This means that the total physical boundary of the compression chamber moves along as the gas is compressed. This is due to the innovation where the sleeve that replaces the cylinder wall is being pushed by the vane to rotate along. At the same time, during the first half of rotation the sleeve pulls the vane out of the slot and during the second half, the vane is pushed back into the slot in the rotor.

3.2.2 Geometry Design

3.2.2.1 Geometry of Compression Concept

Generally, this concept consists of two (2) eccentric circles as shown in Figure

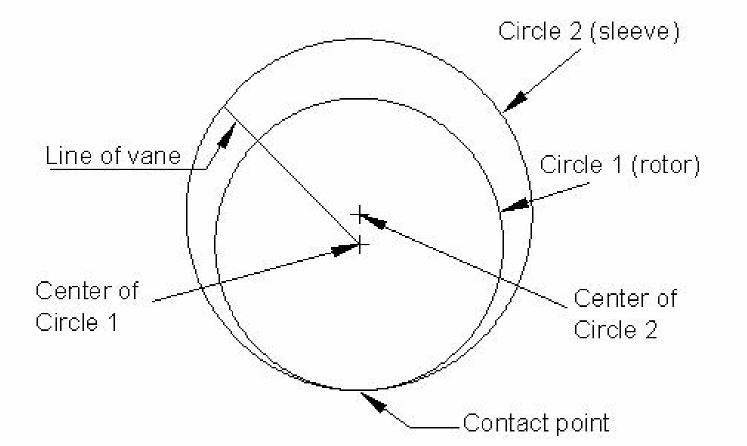


Figure 3.1: Basic geometry of compression concept

Circle 1 function as a rotor and circle 2 as a sleeve with different center points. These circles touch each other only at one point which is called the ‘contact point’. A line from center of circle 1 to any point on circle 2 represents a vane at that particular position of the rotation. Figure 3.2 shows the geometrical relationship of this compressor concept. The area of cde (□cde) is the compressed area which is to be derived in term of other geometrical areas. Angle theta (θ) is the rotation angle of the rotor that must be determined. An analysis has been done to get an expression to relate θ with the swept area. To derive the relationship some expressions are defined with the help of Figure 3.2

$$\mathrm{Sin}\,\theta = \frac{af}{ab}\text{, Thus } af = ab\,\mathrm{Sin}\,\theta$$

Where, $ab = R - r$, thus;

$$\mathrm{Sin}\,\alpha = \frac{af}{ae}\text{, thus } \alpha = \mathrm{Sin}^{-1}\left[\frac{(R-r)\mathrm{Sin}\,\theta}{R}\right]$$

And $ae = R$

$eb = ef - bf$

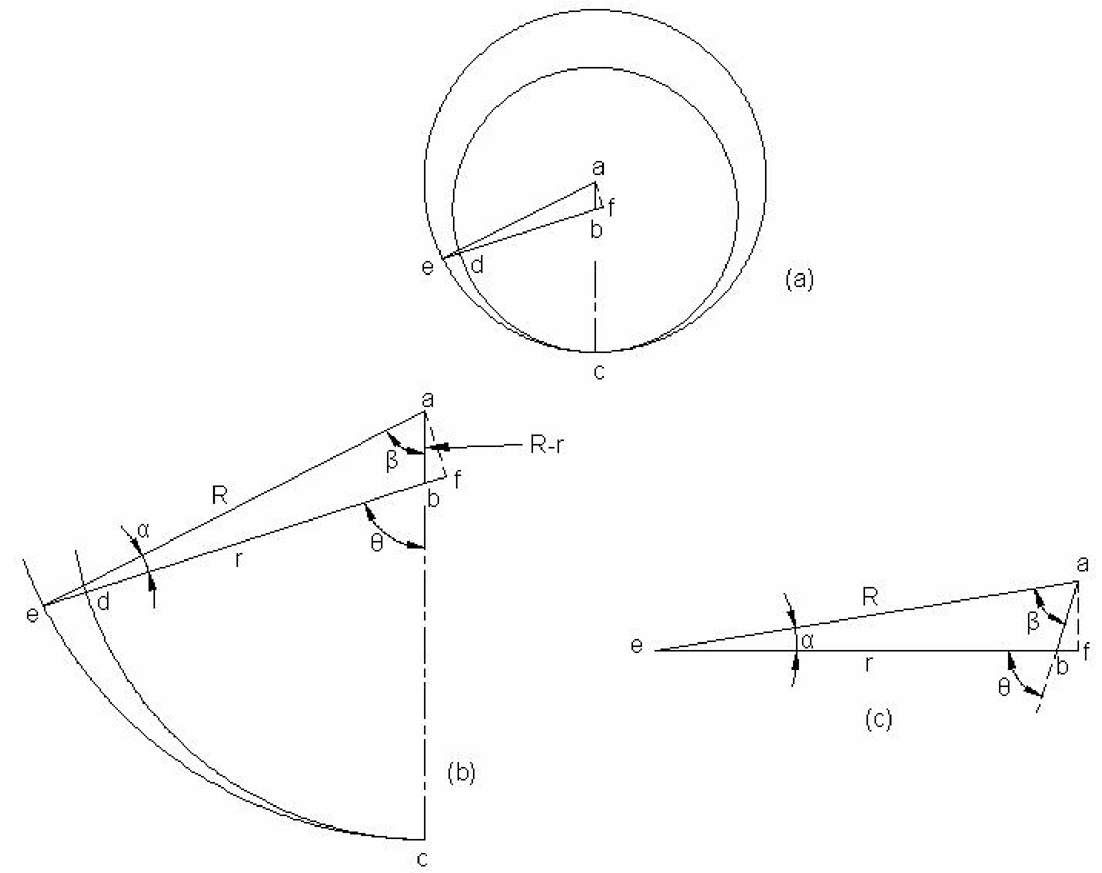


Figure 3.2: Geometry of compressor concept

With

$$ef = R\,\mathrm{Cos}\alpha$$

and $bf = (R - r)\,\mathrm{Cos}\theta$

It can be shown that;

$$eb = R\,\mathrm{Cos}\alpha - (R - r)\,\mathrm{Cos}\theta$$

$$= \mathrm{R\ Cos}\left\{\mathrm{Sin}^{-1}\left[\frac{(R - r)\,\mathrm{Sin}\,\theta}{R}\right]\right\} - (R - r)\,\mathrm{Cos}\theta$$

Here.

$$\beta = \theta - \alpha$$

$$= \theta - \mathrm{Sin}^{-1}\left[\frac{(R - r)\mathrm{Sin}\theta}{R}\right]$$

The area to be determined is Δcde, thus;

$$\Delta cde = \Delta ace - \Delta bcd - \Delta abe$$

Where;

$$\Delta ace = \frac{\beta}{360}\left[\frac{\pi D^2}{4}\right]$$

$$= \left\{\theta - \mathrm{Sin}^{-1}\left[\frac{(R - r)\mathrm{Sin}\,\theta}{R}\right]\right\}\frac{\pi R^2}{360}$$

$$\Delta bcd = \frac{\theta \pi r^2}{360}$$

$$\Delta abe = \frac{1}{2}(af)(ef - bf)\text{, where } ef - bf = eb$$

$$= \frac{1}{2}\left[R\,\mathrm{Cos}\left\{\mathrm{Sin}^{-1}\left[\frac{(R - r)\mathrm{Sin}\theta}{R}\right]\right\} - (R - r)\mathrm{Cos}\theta\right][(R - r)\mathrm{Sin}\theta]$$

All of the equations involved are expressed in term of *R*, *r* and □respectively. The value of □varies from 0□to 360□whereas the values of *R* and *r* are to be specified. Thus, the values of parameters *R* and *r* must be decided to determine other parameters. Design ratio is very important in order to determine the *R* and *r*, based on the published work the recommended value is taken as 0.83 [29]. Thus,

$$\frac{r}{R} = \frac{d}{D} = 0.83 \text{ or } d = 0.83D \tag{4.12}$$

$V = \frac{\pi}{4} D^2 t$, t is the height of rotor and sleeve and is taken equals 20 mm. Thus,

$$6.6 = \frac{\pi}{4}(D^2 - d^2)(2)$$

$$(D^2 - d^2) = 4.2017 \tag{4.13}$$

Substituting $d = 0.83D$ into equation (4.13), thus,

$$D^2 - (0.83D)^2 = 4.2017$$

$$D^2 - (0.6889D^2) = 4.2017$$

$$0.3111D^2 = 4.2017$$

$$D = 36.75 \text{ mm}$$

Substituting $D = 36.75$ mm into equation (3.12),

$$d = 0.83D$$
$$= 0.83(36.75)$$
$$= 30.5 \text{ mm.}$$

But, in this design the suction port causes a decrease in the induced volume. Some gas is pushed out as the vane rotates from *c* to *e* with D = 36.75 and d = 30.5 mm, using equation (3.8) the actual swept volume is 6.6̃ (□*cde* □2) cm^3. To compensate for this loss of gas and at the same time maintaining the optimum r/*R* ratio of 0.83 both radii have to be increased. Again, using AutoCAD software and through trial and error, *d* was obtained equals to 32 mm and *D* equals to 38.5 mm. These dimensions are shown in Figure 4.3.

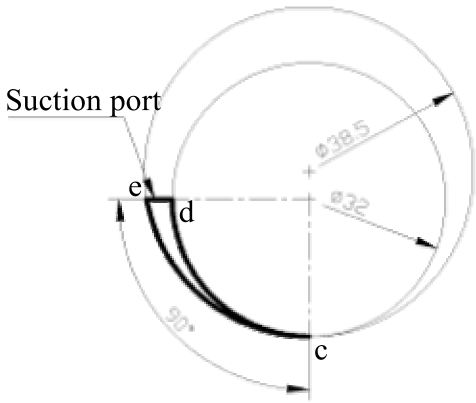


Figure 3.3: Suction port position of new rotary compressor

For further check, with the new values of *d* and *D* respectively, the actual swept volume can be calculated using equation 4.8 as follows:

$$V = \frac{\pi}{4} D^2 t$$
$$= \frac{\pi}{4}\left(3.85^2 - 3.2^2\right)(20)$$
$$= 7.1982 \ \text{cm}^3, \text{ is total swept volume of the compressor.}$$

Referring to Figure 3.3, compression is shown to begin at theta (θ) equals to 90°. From equation (3.8), the effective swept volume can be calculated. Thus, from figure 3.2;

$$\Delta cde = \Delta ace - \Delta bcd - \Delta abe$$
$$\Delta ace = \left\{\theta - \text{Sin}^{-1}\left[\frac{(R-r)\text{Sin}\,\theta}{R}\right]\right\}\frac{\pi R^2}{360}$$
$$= \left\{90 - \text{Sin}^{-1}\left[\frac{(1.925-1.6)\ \text{Sin}\ 90}{1.925}\right]\right\}\frac{\pi(1.925)^2}{360}$$
$$= \frac{(80.28)(1.925)^2\pi}{360}$$
$$= 2.5961 \ \text{cm}^2$$

$$\Delta cde = \Delta ace - \Delta bcd - \Delta abe$$

$$\Delta ace = \left\{ \theta - \mathrm{Sin}^{-1}\left[\frac{(R-r)\mathrm{Sin}\,\theta}{R}\right]\right\}\frac{\pi R^2}{360}$$

$$= \left\{ 90 - \mathrm{Sin}^{-1}\left[\frac{(1.925-1.6)\,\mathrm{Sin}\,90}{1.925}\right]\right\}\frac{\pi(1.925)^2}{360}$$

$$= \frac{(80.28)(1.925)^2\pi}{360}$$

$$= 2.5961\ \mathrm{cm}^2$$

So, the volume of *cde* is $0.27718(2) = 0.55436\ \mathrm{cm}^3$ and this is the amount of gas that is being pushed out. The actual swept volume (V_S) of compressor is;

$$V_S = 7.19817 - 0.55436$$

$$= 6.644\ \mathrm{cm}^3$$

This approximately equals to the swept volume of the existing reciprocating compressor.

3.2.2.2 Geometry of Vane

The vane has a round head and straight body and is respectively assembled into an open round slot on the inner part of the sleeve and in a radial slot on the rotor such as illustrated in Figure 3.4.Therefore, the vane, sleeve and rotor are dynamically linked and for the kinematics to be possible, the opening angle of the slot in the sleeve has to be correct to allow the vane a certain degree of swing and at the same time preventing the vane from being detached. Determination of this angle is described in section 3.2.2.4.

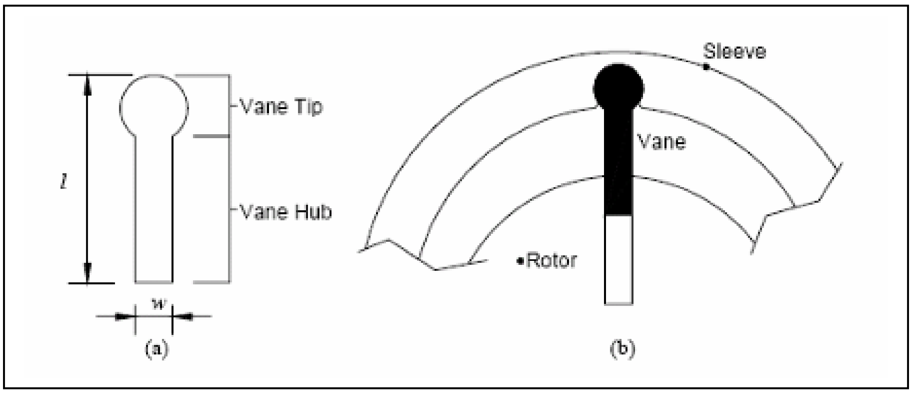


Figure 3.4: Profile of vane design

From force analysis of the vane thickness, *w*, obtained is very small and does not provide enough hydraulic seal to prevent internal leakage at both sides of the vane. Examination of existing rolling piston rotary compressor of the same capacity, a vane thickness of 2.5mm was believed to be equally suitable and therefore adopted. A vane tip of diameter 4.5 mm is found sufficient for a circular slot of same diameter to be cut in the sleeve to operate safely against a design pressure of 30 bar. The vane length (*l*) is determined by the eccentricity between rotor and sleeve. The length must be greater than the eccentricity to maintain enough part the vane to remain in the slot such as illustrated in Figure 3.4 (b). When the vane is fully pushed in the rotor slot, the provision of a rounded space allows enough clearance for lubrication.

3.2.2.3 Geometry of Rotor

The rotor is the driving component in this compressor concept. Power is transmitted from a motor to the rotor through a shaft. Referring to Figure 4.5 a slot is cut radially in the rotor to facilitate the radial movement of the vane while simultaneously transferring torque to the vane and through the vane to the sleeve. The length of the slot is determined when it is at 0° of rotation at which the entire vane hub is in the slot, leaving a small clearance which is occupied by lubrication oil. The maximum length of vane that enters into the slot is 10 mm, thus the length of vane slot was designed to be 10.5 mm with a round shape at the end of slot. The shaft dimension was decided equal to 10 mm diameters. This is strong enough to take the torque during compression. To simplify machining process, the inner end of the vane slot is designed not to touch the shaft.

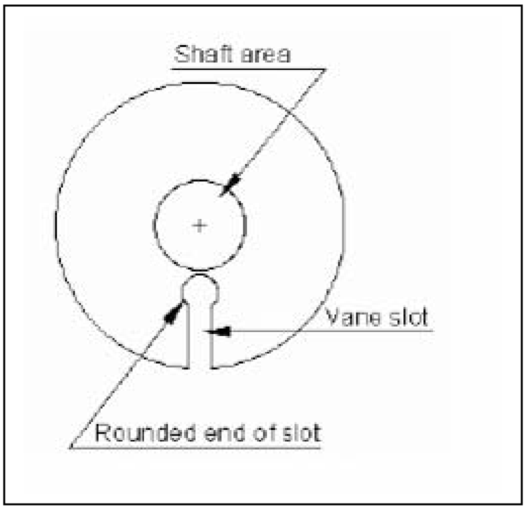


Figure 3.5: Rotor profile design

3.2.2.4 Geometry of Sleeve

Referring to Figure 3.4 (b) again, the sleeve is designed with a slot having a similar profile with that of the vane tip. The vane slot of sleeve must have opening angle to allow the vane to swing smoothly. The vane swinging angle on the sleeve was obtained by trial and error using AutoCAD software by rotating the mechanism as shown in Figure 1.7 in section 1.3. Using the procedure a vane angle of 91was selected. At the opening of the slot, fillets are cut on each side to minimise the risk of the edges being damaged during operation. The outer diameter of sleeve was determined by force analysis but the actual value was decided to match the standard size of the needle roller bearing. The suitable bearing selected has an inner diameter of 50 mm and a height of 20 mm. Thus, the chosen outside diameter of sleeve is 50 mm.

3.2.3 Discharge Angle Calculation

As described previously, compression starts at pressure p_1 when the vane has just rotated over the suction port. At certain angle of rotation, the pressure reaches p_2. It is at this point that the gas must be discharged. The theoretical discharge angle can be calculated using geometrical relationship and using AutoCAD. Both of these methods are based on the estimation of discharge volume V_2 from the following equation.

$$\frac{V_1}{V_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

where V_1 is the actual swept volume equals 6.64 cm^3 and p_1 and p_2 are specified values. The polytropic index (n) was calculated using equation 3.6

$$\frac{V_1}{V_2} = \left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

where, T_1, p_2 and p_1 are actual values measured from experiment. The value of T_2 was estimated from isentropic process of compression ($s_1 = s_2$). The data of T_1, p_1, and p2 were taken from a preliminary run of the experiment at which the freezer compartment temperature was constant at -15°C as below:

$T_1 = 26.4^{o}C$ $\quad$ $p_2 = 9.8$ Bar G $\quad$ $p_1 = 0$ Bar G

Therefore the values of s_1 and s_2 were estimated equal to 1.036 kJ/kg K. Based on NIST REFPROP software from ASHRAE, the value of T_2 is 105.2^{0}C. Thus, using equation 3.6, the polytropic compression index (n) is obtained equal to 1.11. From equation 3.7, the discharge volume (V_2) can be calculated as equal to 0.7866 cm^3.The theoretical discharge angle was determined based on the area of A_2 which is equal to 0.3933 cm^2. By using AutoCAD, it was found that the corresponding discharge angle is about 258^{o}C.

3.2.4 Design of Compression Component

The components that creates the compression chamber such as rotor, vane and sleeve were designed based on the geometrical dimension determined earlier. Height of these components are equal to 20 mm with tolerance of 0 mm to -10 µm. The rotor was designed as one piece with shaft and dimension of the shaft was matched with the needle roller bearing [54]. The inner diameter (ID) of bearing 1 is 8 mm and ID of bearing 2 is 10 mm. The tolerance for radius of rotor must be determined by considering the maximum tolerance of contact point between rotor and sleeve. Since the tolerance of contact point was decided as 7.5 □m, the rotor tolerance was obtained as –5 µm to 0 µm in diameter. All of above discussion is illustrated in Figure 3.6.

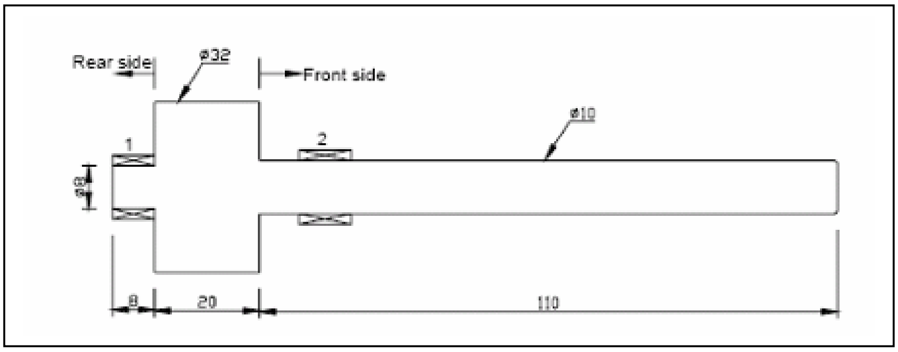


Figure 3.6: Detail of rotor design

As mentioned earlier, the sleeve needs to be supported with a bearing which has a 50 mm ID. A needle roller bearing of 50 mm ID, SKF 5020 is found suitable. This bearing has a 58 mm outer diameter. The tolerance of sleeve inner diameter should be referred to the rotor tolerance and sleeve/rotor contact point clearance. Thus, the tolerance of sleeve inner diameter was obtained as 0 µm to +10 µm. The vane was designed based on the geometry that has been discussed with 20 mm height. The tolerance of vane height is equal to the tolerance of rotor and sleeve height whereas the tolerance of vane width was obtained as –12 µm to 0 µm and vane slot at rotor was obtained as 0µm to +10 µm. The cylinder block was designed with center bore to insert the needle roller bearing HK 5020 SKF and all compression components. Height of cylinder block was designed equal to 20 mm with +5µm tolerance. Figure 4.7 shows the disposition of bearing, sleeve, rotor and vane into cylinder block.

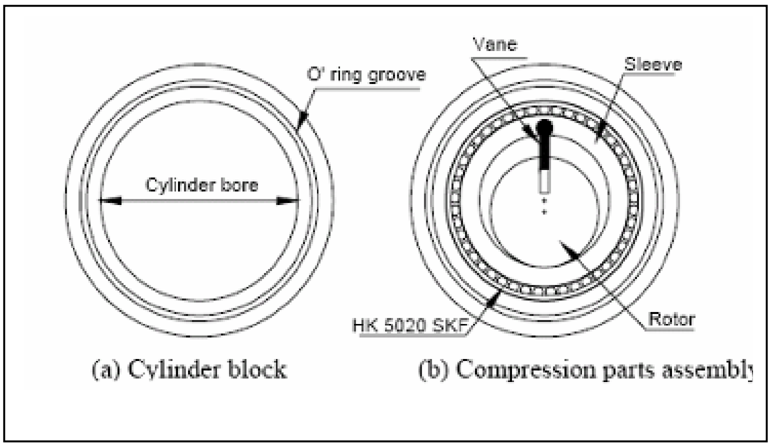


Figure 3.7: Compression parts assembly into cylinder block

End plates form an enclosure to the compression chamber as well as to support and to alignment the shaft and to provide smooth rotation by installing a bearing on each end plate. Referring to Figure 4.8, at the front end plate, it was assembled with HK 1010 needle roller bearing (2) and oil seal (3) to prevent the high-pressure gas leaking out of the compression chamber. Needle roller bearing HK 0808 (1) was assembled at rear end plate. The passages of suction and discharge are through the front end plate and rear end plate respectively.

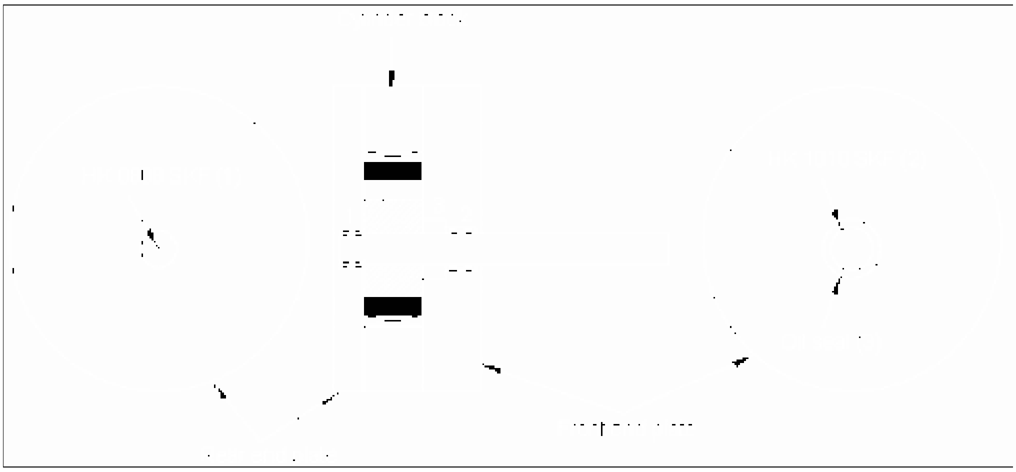


Figure 3.8: Detail of end plate design

CHAPTER 4

CONCLUSION AND SUGGESTIONS

4.1 Conclusion

A new idea of a single vane rotating sleeve rotary compressor was conceived. The concept development and literature study carried out in this report is a preliminary step in design and development process of the new concept. The patent for the invention has been duly filed. The reference patent number is PI 20014899 entitled Single Vane Rotary Compressor under Malaysian Patent Office.

The new concept has attracted investors from local companies for commercialization activities. One such company is Thermocompressor Engineering Sdn Bhd who is keen in developing the concept further and commercializes it for the small portable air conditioning market. Due to the concept advantage, it is possible to develop a compressor which is smaller in size yet producing the same cooling capacity compared to other concept within the same class. The project had received research grant from the government under MTDC with a total amount of RM 3.4 million with half of the amount is provided by the company. Currently the company is allocating the required amount of RM 1.7 million before the project can progress further.

Works are already started in developing a working prototype for the compressor as well as commissioning a test system for measuring the performance of the compressor. The research work will be continued on other research grant (74522) as the current grant is unable to support the research any further.

4.2 Suggestions

Every new invention is followed by further development work. The short few months spent was felt not enough to develop a working prototype whereby the new rotary compressor can be commercialized and accepted by the open market. The following suggestion are proposed to be considered to be taken so as the innovation will lead to a successful application of the new invention on to the refrigeration plants like domestic refrigerator and split unit air-conditioning system.

1) A comprehensive research and development work should be done in the near future to fully utilize the advantage of the new compressor in the open market.
2) The development work should include in concept improvement as well as a through engineering analysis covering kinematics and dynamics of the concept to final product durability and fabrication method.
3) Since the new concept is universal, work should also cover other applications such as automobile air conditioning or hydraulic pumps and vacuums.
4) A test rig should be developed to test the performance of the new compressor and a fair comparison should be made to that of existing compressor readily available in the market

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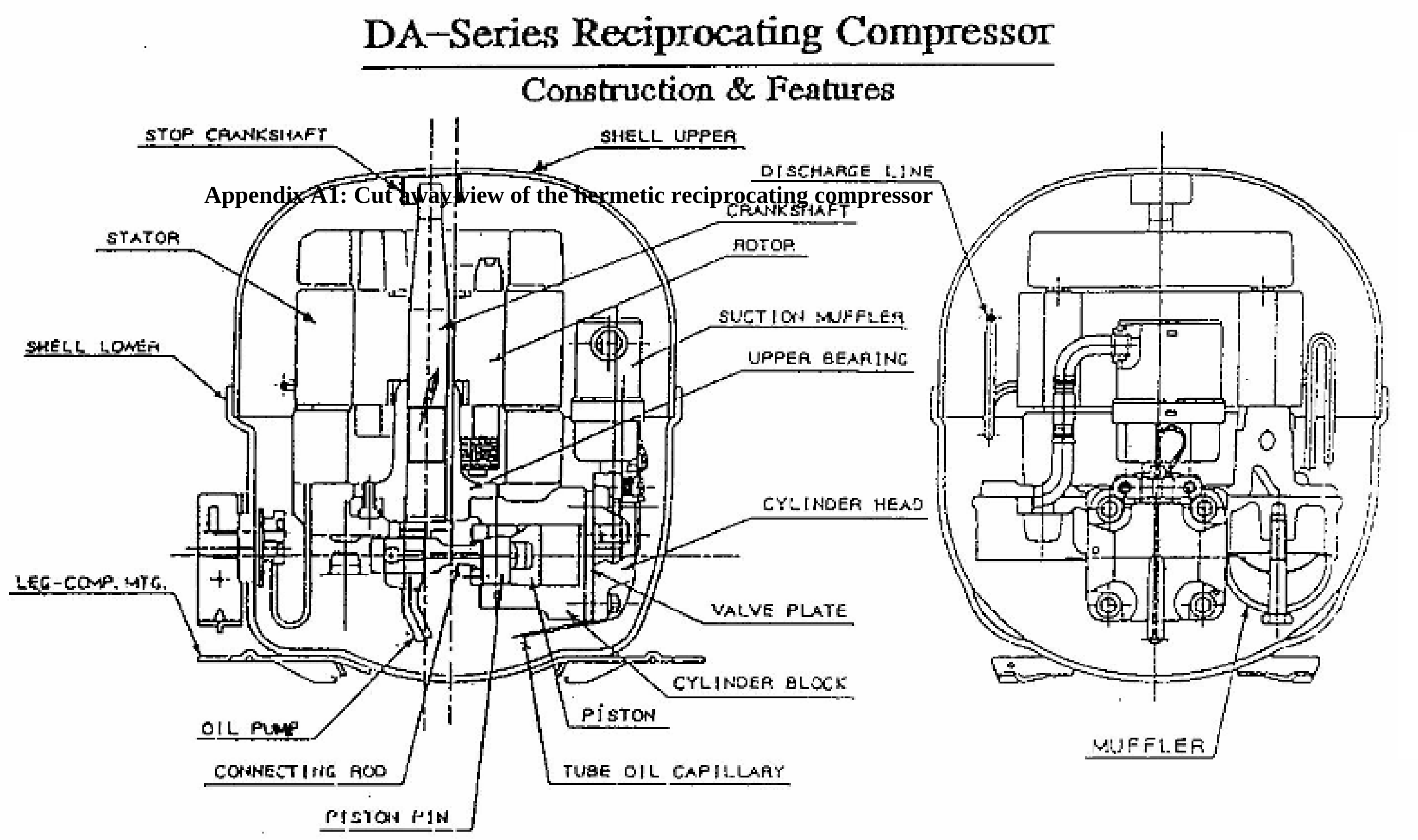
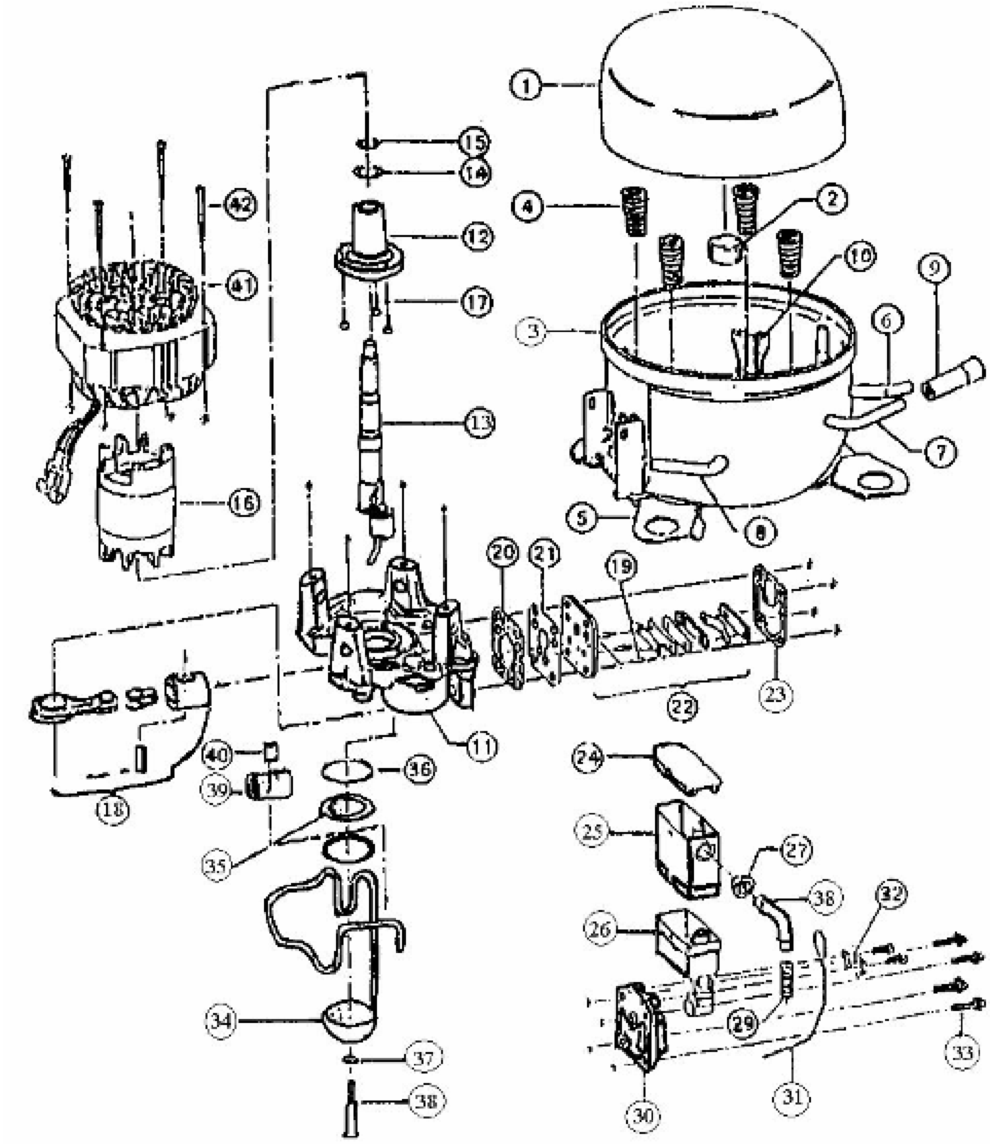
Appendix A1: Cut away view of the hermetic reciprocating compressor



Appendix A2: Exploded view of the hermetic reciprocating compressor



1	Shell-Upper	15	Thrust Bearing B	29	Joint Spring
2	Stop-Crankshaft	16	Rotor	30	Cylinder Head
3	Shell-Lower	17	Cap Screw	31	Tube-Oil Capillary
4	Suspansion Spring	18	Piston & Con. Rod Assy.	32	Support
5	Leg-Compressor Mount	19	Pin-Valve Guide	33	Cap Screw Hex. Flange
6	Tube-Suction	20	Gasket-Valve Plate	34	Muffler Cover / Disch. Line Assy.
7	Tube-Discharge	21	Reed-Suction Valve	35	Baffle-Discharge Muffler
8	Tube-Process	22	Valve Plate Assy.	36	Gasker Muffler
9	Cap Tube	23	Gasket-Cylinder Head	37	Gasket Washer
10	Stay Assy	24	Suction Muffler A	38	Hex. Head Bolt
11	Block Compressor	25	Suction Muffler B	39	Damping Strap
12	Upper Bearing	26	Suction Muffler C	40	Clip Damping
13	Crankshaft	27	Damper Collar	41	Stator Assembly
14	Thrust Washer A	28	Insert Tube	42	Screw Hex. Head (Stator)

Refer to the manual of this compressor, there are 42 parts involved in the assembly and to ensure that the compressor is able working properly. Each of parts that involved having distinctive function such as described below;

1 & 3 – Upper and Lower Shell

These shells are welded together to form a hermetic seal.

2 – Stop Crankshaft

This is spot welded to the upper shell to prevent the toppling of pump assembly especially during transportation of the compressor. It also prevents the pump assembly dislocated from the spring.

4 – Suspension Spring

To isolate the vibration of the pump assembly from the shell and to provide free movement of the pump assembly during compressor starting and stopping.

5 – Leg Mounting

To facilitate the mounting of the compressor to the cabinet.

6, 7 & 8 – Tube Suction, Tube Discharge and Tube Process

Tube suction and tube discharge are to facilitate the connection of tube suction to the evaporator and tube discharge to the condenser of the cabinet. Tube process is to facilitate the charging of refrigerant to the system, vacuuming of the system, as well as the charging to oil to the compressor.

9 – Cap Tube

To seal off tube process, discharge and suction from foreign particles, dust, moisture and others.

10 - Stay

To facilitate the suspension springs mounting.

11 – Block

Attachment of upper bearing, discharge muffler, stator and valve system.

12 – Upper Bearing

Act as a bearing support for the rotating mechanism.

14 & 15 – Thrust Washer

To share the weight of the rotor acting on the upper bearing surface.

20 – Gasket Valve Plate

With 5 different of thickness to adjust the top clearance to the minimal so as to eliminate top noise and maximize compressor efficiency.

21 – Reed Suction Valve

During the suction process, it allows the refrigerant to enter the cylinder. During the compression and discharge process, it prevents the refrigerant from escaping into the suction chamber.

22 – Valve Plate Assembly

Discharge Reed – During the suction process, it prevents the discharge gas from returning to the cylinder, and during the discharge process, it allows the refrigerant to be discharged to the discharge chamber, thus serving as a control valve.

Valve Plate – Acts as a base plate for the mounting of discharge reeds, it has discharge and suction holes, discharge and suction passes and guide pin holes on it.

23 – Gasket Cylinder Head

To prevent leakage of discharge gas and leakages between suction and discharge chamber.

24, 25 & 26 – Suction Muffler

To reduce the temperature of suction gas so as to improve compressor efficiency and reduce noise.

27 – Damper Collar

To prevent dislocation of the insert tube with the inlet of the suction muffler, and to absorb the vibration of the pump assembly.

29 – Joint Spring

To absorb the lateral vibrational movement of the pump assembly

30 – Cylinder Head

To seal the valve plate assembly and to isolate the suction and discharge chamber.

31 – Tube Oil Capillary

To supply oil to the cylinder serving as lubricant and to reduce the leakage of high pressure gas.

34 – Muffler Cover & Discharge Line Assembly

The muffler cover is fastened to the block forming the discharge muffler, which reduces the discharge gas pulsation and noise. The discharge line acts as a passage between the discharge muffler and the tube discharge.

35 – Baffle Discharge Muffler

It reduces discharge gas pulsation and noise.

39 – Damping Strap

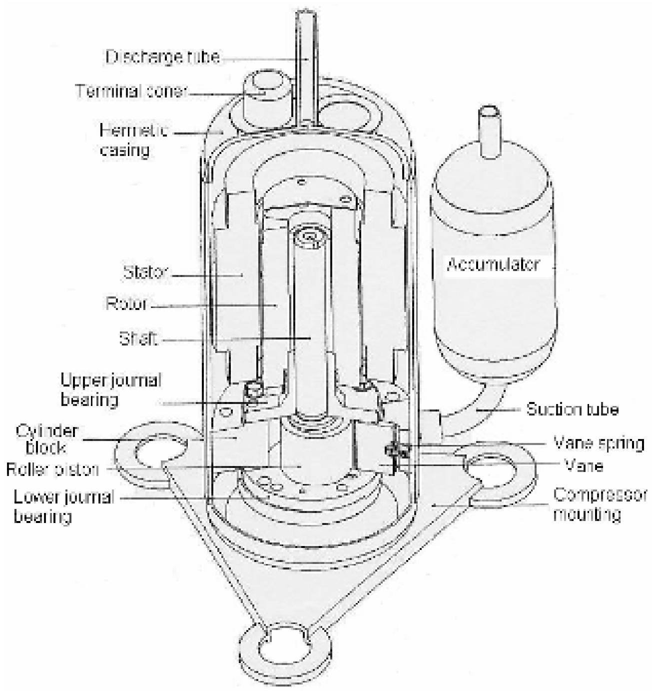
To reduce the vibration on the discharge line.

Lubricating Oil

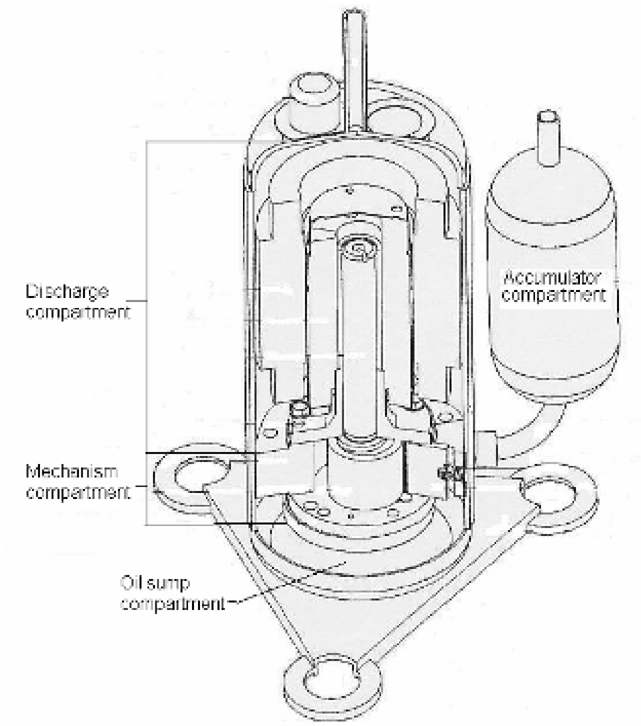
For lubrication, cooling, and sealing purpose. Lubrication is required to reduce friction and wear.

APPENDIX B

Appendix B1: Section view of rolling piston rotary compressor



Appendix B2: Compartment boundaries of rolling piston rotary compressor

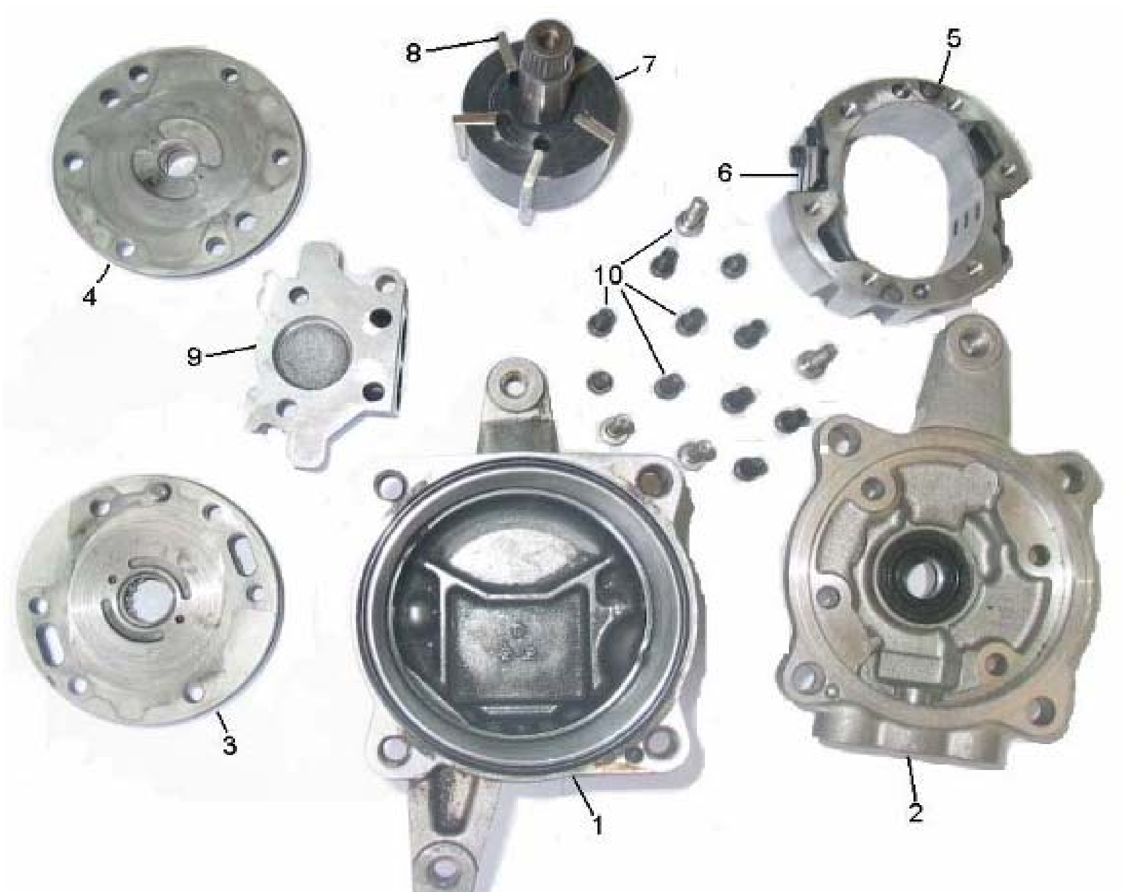


APPENDIX C

Appendix C1: Dismantled component of Patco compressor

Legend:

1- Casing
2- Front cover Casing
3- Cylinder block
4- Rear end plate
5- Front end plate
6- Rotor and shaft
7- Vane
8- Oil Separator



Appendix C2: Dismantled component of Denso compressor

Legend:

1- Casing
2- Front cover casing
3- Front end plate
4- Rear end plate
5- Cylinder block